HYDRAULIC OPERATION OF MACHINES.

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Introduction.

THERE is certainly nothing new in the simpler applications of fluid or hydraulic power, as it is usually termed, for it has been used for very many years to energise presses, cranes, jacks, lifts, and later, the hydraulic brakes of automobiles.

These and other applications are based upon a fundamental law of fluids, namely, Pascal's Principle, which states: If the pressure at any point of a fluid, at rest, be increased, that pressure is transmitted equally in all directions, if the fluid be incompressible. Thus if a piston is pushed downwards, the pressure it exerts on the surface of the fluid is transmitted equally to the sides and bottom of the cylinder.

To exert a large force by applying a small one, Pascal's law is put into operation in a hydraulic press. A large cylinder and piston are connected to a small cylinder and piston by a pipe. The system is filled with oil, and assuming that the area of the small piston is "a," the pressure exerted by the small piston to be p and the area of the large piston A, then the total pressure, P of the large piston on the work, W, is by Pascal's Principle, p times the ratio of area A to area "a," or $P = p \frac{A}{a}$

If "a" were 1 sq. in. in area and "A" 10 sq. in. in area with the pressure p of the small piston 1 lb., the pressure of the large piston on the work would be 10 lb.

In ordinary hydraulic installations with accumulators in circuit, the maximum power is used at all times irrespective of whether it is required, but in the more recent developments utilising a variable delivery pump to each machine, no power is wasted as the pressure of the oil and the power absorbed is only that of the resistance being overcome.

Fluid power may be compared to the voltage of an electric generator, the discharge similarly is related to the rate of current flow or amperage. Instead of two wires, two pipes are used to transmit hydraulic transmsission, the suction and discharge pipes corresponding

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to the negative and positive leads of an electric generator. Due to its similarity to electric current, fluid power may be transmitted over long distances, over, through, or around obstacles into positions inaccessible by belts, chains, gears, or other means of power transmission.

The resemblance to electric power can be carried still further, for fluid power may be interrupted, reversed, or varied in intensity almost instantaneously, without stopping or changing the speed of the prime mover or stopping the pump unit.

Presses constitute a large and primary field to which hydraulic operation has been traditionally applied, but the development of the variable delivery pump has widened the sphere of operation so that it now includes practically every type of machine in some form.

To consider a typical press arrangement, oil can be delivered by the pump through an operating valve to either side of the piston to raise or lower it. Thousands of industrial applications are merely slight departures from this simple application.

A piston can be made to rise or fall by valve control in an unlimited diversity of ways. Thus, if the horizontal distance represents the time required for a complete cycle of operation and the vertical travel of the ram is limited, say " a " to " b," it is possible by hydraulic means for the plunger to start at " a " and terminate at " b " in any manner that the designer desires. This may be regarded as typical of the manner that fluid may be used to actuate definite machines and mechanical functions.

Hydraulic Circuits.

In general a hydraulic transmission unit may work on either an open or a closed circuit. With an open circuit the pump draws oil from the tank, to which it is returned via the exhaust pipe after passing through the valves and cylinder. One arrangement can comprise simple constant speed operation of a single cylinder with gravity return for the ram. The valve is so arranged that oil passes direct to the cylinder for the vertical ram traverse. the exhaust line meanwhile being closed. Rotation of the valve cuts off the oil supply from the pump and opens the exhaust line for the gravity return of the ram.

A double acting cylinder will give unequal speeds in either direction unless fitted with a tail rod. Control is by a simple reverse valve. If equal speeds in both directions are required, a constant pressure is maintained on the small end while the large end is opened to pressure or exhausted through the valve. It is necessary that the cylinder area be equal to twice the area of the piston rod to give equal speeds in either direction.

The open circuit arrangement is generally preferable to the closed circuit, being simpler and allowing a free oil circulation from and

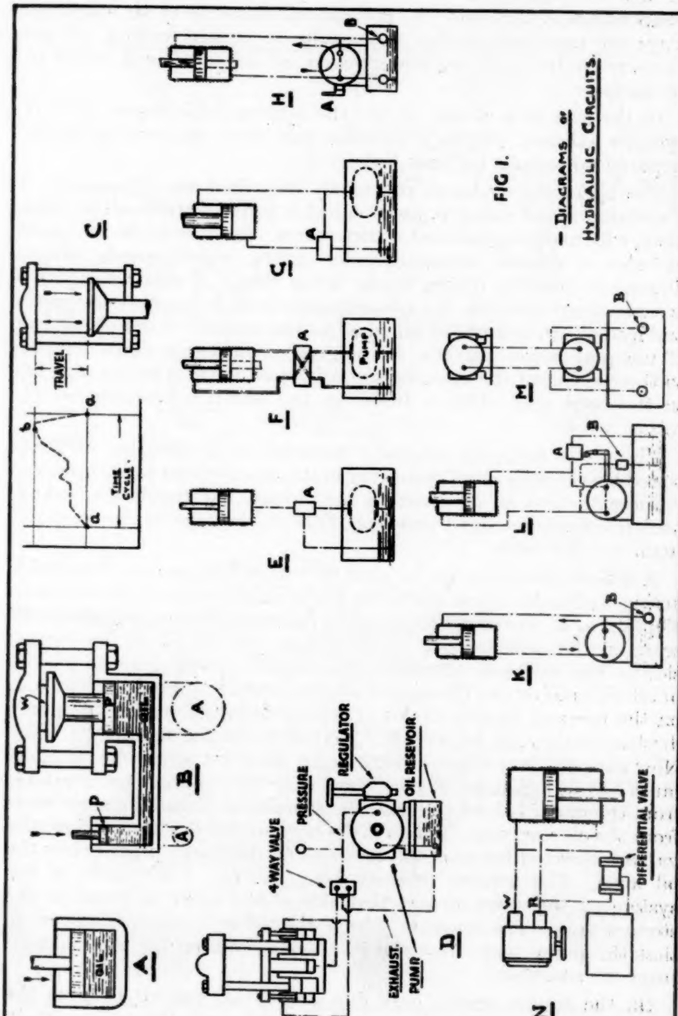


FIG I.

— DIAGRAMS OF HYDRAULIC CIRCUITS.

to the reservoir. Due to the large quantity of oil in circuit the temperature is kept low. To prevent the entrance of air which destroys the incompressibility and results in erratic feeding, all pipe joints must be tight and the exhaust oil delivered well below the oil surface.

In the case of a closed circuit, the suction oil is drawn from the cylinder exhaust, although make-up lines from the tank are always required to supply for any leakage.

The operating cycles as previously described are obtainable. If a variable speed pump is used, suitable lever control can be fitted along with make-up and non-return valves. Another cycle obtainable includes a double acting cylinder giving equal speeds in both directions, but the return stroke being idle. A non-return valve is set in the return line at a pressure just enough to return the piston and by-pass an amount of oil equal to the volume of the piston rod. If unequal power traverse is required, a make-up valve supplies to the rod end of the cylinder and a by-pass valve is in the pipe line to the large end. This is lifted by the pump mechanism on the return stroke.

The pump units are generally mounted in or upon the oil tanks to reduce the number of mounting units, an overhead tank, however, reduces the risk of air entering the circuit, but requires a leakage tank from which a small pump conveys the leakage to the overhead main supply tank.

A differential valve can be used to compensate or relieve a double acting hydraulic sytem from the differential volumes present in it. The valve is so constructed that a floating, double seated spindle seats on the pressure side of the system and opens on the suction side to the reservoir. Suppose in a double acting press 15 gallons of oil are used on the downward working stroke and only five gallons on the upward, return stroke. On the downward stroke the pump discharges through flange " W " and sucks oil through flange " R." Now suppose that the ram is to begin its down stroke for which it must have 15 gallons of oil. Since only five gallons are available from the under side of the ram, the remaining 10 gallons must come from the oil reservoir. This is easily accomplished with a differential valve connected between the suction and discharge pipes below the oil level. The greater pressure being on the " W " side of the system on the down stroke, that side of the valve is closed to the storage tank. The opposite side of the valve is open, however, so that the pump may draw the 10 gallons required for the make-oil from the reservoir.

On the return stroke, only five gallons are needed to push the ram upward, yet 15 gallons are available and the excess of 10 gallons must be disposed of. This time, greater pressure is on the

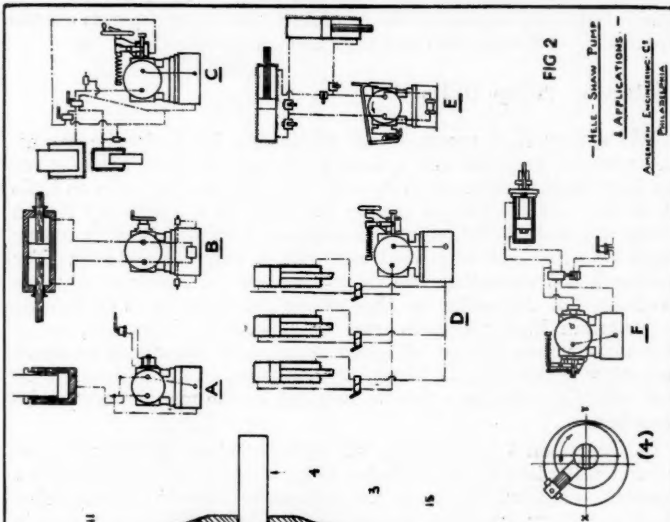


FIG 2
— HELE - SHAW PUMP
& APPLICATIONS —
AMERICAN ENGINEERING Co
PHILADELPHIA
A
B
C
D
E
F
(1)
(2)
(3)
(4)
STROKE

" R " side of the system, closing off that side to the reservoir. The opposite side is open, and the surplus 10 gallons which the pump does not require pass through the differential valve to the reservoir. The selection of a proper size of differential valve depends on the volumetric difference and the time element involved.

Hydraulic Pump Units.

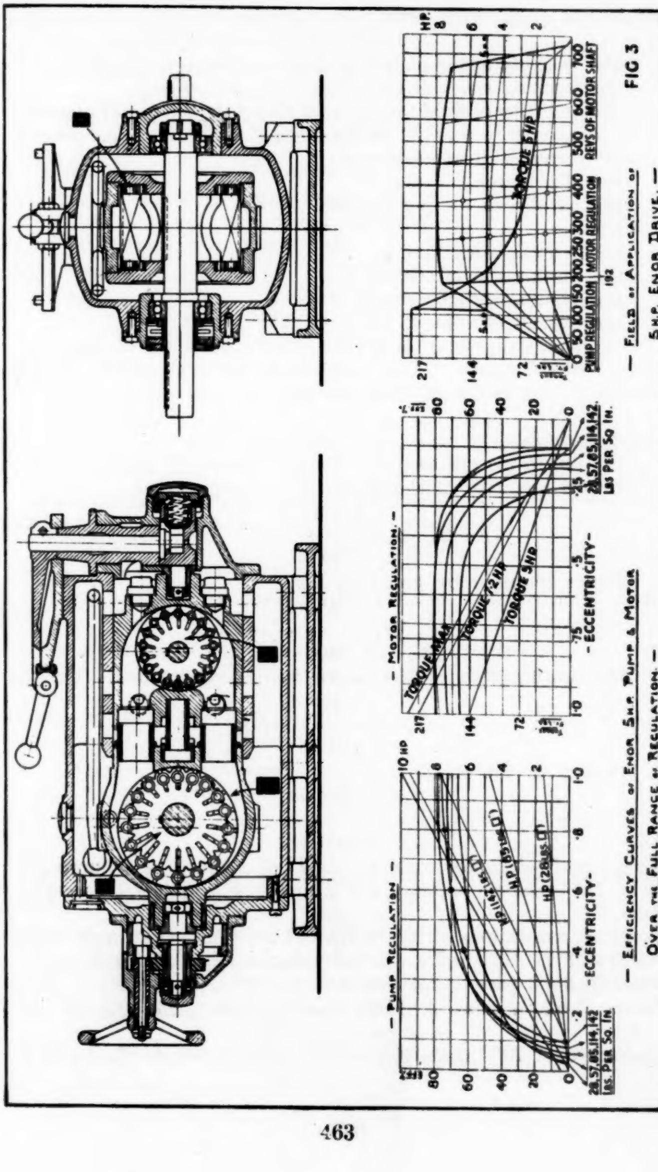
While very high pressures are obtainable by hydraulic means, the pressure required for actuating driving or feed motions for machine tools is comparatively low, actually ranging from as little as 75 lb. to 1,200 lb. per square inch. As it is necessary that a pump will deliver oil free from pulsations, multiple piston or rotary vane type are generally used and give a variable delivery. For pressures not exceeding 400 lb. per square inch gear pumps are available but depending as they do on line contact, it is difficult to maintain high pressures over long periods. Moreover, being constant delivery pumps, all speed variation is dependent on valve control, whereas with the piston or vane type, an infinitely variable and reversible discharge can be obtained in the manner now to be described.

In the piston type of pump the cylinder block houses seven or more radial pistons, the cylinder being driven around a variable eccentric crank which causes the pistons to travel in the cylinder block and draw up oil on one side of the stroke, and to return it under pressure on the opposite side. When the eccentrics are rotated to cancel out, no motion of the pistons within the cylinder block takes place and the oil flow ceases. By moving the eccentrics from one side to the other, a reversal of flow takes place. In some cases the cylinder block itself is the moving component, being swung from side to side to give forward or reverse oil delivery. Yet another development is a tilting block arrangement on the lines of the Williams—Janny first produced in 1907 and now constructed in an improved form by Messrs. Variable Speed Gear. The function of the tilting block is to control the stroke of the pistons and thereby the oil flow.

With the second type, that is the rotary vane, again great accuracy of construction is essential but an advantage claimed is that the oil pressure does not act radially but nearly tangential to the rotor.

On the Enor pump the oil flow is regulated by varying the eccentricity between the axis of the housing and that of the blade rotor, this is a straight line motion. Wear is reduced by guiding the blades by rollers in circular race tracts which take up centrifugal force.

As the method of calculating output, etc., for this type of pump appears little known, formulae and tables are given herewith.


— Pump Regulation —
— Motor Regulation —
- ECCENTRICITY -
- ECCENTRICITY -
TORQUE MAX
TORQUE 7.2 HP
TORQUE 5 HP
TORQUE 5 HP
28.57.85.114.142
LBS. PER SQ. IN.
PUMP REGULATION
MOTOR REGULATION
REVS OF MOTOR SHAFT
H.P.
— Efficiency Curves of Enor 5 H.P. Pump & Motor
Over the Full Range of Regulation. —
— Field of Application of
5 H.P. Enor Drive. —
FIG 3

Calculations of the Enor Unit.

Let—

q = the working space of the unit in c.c. per revolution.
Q = the working space of the unit in c.c. per second.
F = the quantity of fluid moved, and therefore the useful quantity of the pump, and oil consumption of motor in c.c. per second.
P = oil pressure in atmospheres.
s = the slip factor.
S = the oil slip in c.c per second.
n = r.p.m.
N = mechanical work in h.p.
ηm = the mechanical efficiency.
ηv = the volumetric efficiency.
η = the overall efficiency.

For a given value of N or F, the other values can be found by means of the formulae. The next procedure is to select suitable values from the tables of N, n, and Q.

General Formulae.

(1) Oil slip in c.c per second. $S=(p+0.5)s$.

(2) Useful quantity of the pump in c.c. per second,

$$F=Q_1-S_1=\frac{q_1\times n_1}{60}-S_1.$$

(3) Power required by pump in h.p.

$$N_1=\frac{Q_1\times P}{7500\times\eta m_1}.$$

(4) Oil consumption of the motor in c.c. per second.

$$F=Q_2+S_2=\frac{q_2\times n_2}{60}+S_2$$

(5) Output of the motor h.p.

$$N_2=\frac{Q_2\times P\times\eta m_2}{7500}.$$

Example of calculation. Enor unit *JV*-3, consists of a pump *J*-3 with a 2/4 eccentric setting, and a motor *V*-5 with 4/4 eccentric setting.

Constant revolutions of the pump=1100 r.p.m. Oil temperature 60°C. (140°F.). Calculations for oil pressure of 10 atmospheres :—

From Table 3 (size 3, 2/4 setting). $q=380$ c.c.

From Table 4 (size 3, 1100 r.p.m.). $\eta m_1=.95$. $s_1=33$ for temperature of 70°C. (158°F.), or $33\times\frac{60}{70}=28$ for 60°C. (140°F.).

Oil slip in c.c. per second (Formula 1).

$S_1=(p+0.5)s=(10+0.5)28=294$ c.c.

Useful quantity of the pump per second (Formula 2).

$$F=\frac{q_1\times n_1}{60}-S=\frac{380\times 1100}{60}-294=6686 \text{ c.c.}$$

Power Requirement (Formula 3).

$$N_1=\frac{Q_1\times P}{7500\times \eta m_1}=\frac{6980\times 10}{7500\times 0.95}=9.8 \text{ h.p.}$$

Motor V5 (see table).

Settings	4/4	3/4	2/4	1/3	1/4
q_2 =	2222	1666	1111	741	555 c.c.
ηm_2 =	0.91	0.93	0.93	0.91	0.84

$s_2=50$ at 70°C. or 43 at 60°C.

(1) Slip per second $S_2=(p+0.5)s=(10+0.5)43=452$ c.c.

(4) Amount of oil the motor must actually use.

$Q_2=F-S=6686$ (delivered from pump to motor)—452 (slip) $=6234$ c.c.

With the same formula can be obtained (for the corresponding settings) the number of r.p.m. of the motor end.

Settings.	4/4	3/4	2/4	1/3	1/4
$n_2=\frac{60\times Q_2}{q_2}=\frac{60\times 6234}{q_2}=$	168	224	336	505	673

Space decreased, pressure and quantity remaining the same, velocity must increase, and r.p.m. increase due to smaller opening.

Output of the Motor. (5).

$$N_2=\frac{Q_2\times p\times \eta m_2}{7500}=\frac{6234\times 10\times \eta m_2}{7500} \qquad \eta m_2=.91, \text{ etc.}$$

Setting	4/4	3/4	2/4	1/3	1/4
H.P.	7.57	7.73	7.73	7.57	6.98

H.P. remains approximately constant, because gain in velocity equals loss in static pressure.

Calculations of the Pressure and Efficiency for 5 h.p.

The pressure p falls approximately, in relation to the desired output, round about 6.5 atmospheres. In these circumstances.

$S_1=(p\times 0.5)s_1=(6.5+0.5)28=196$ c.c.

$S_2=$ $(6.5+0.5)43=301$ c.c.

$Q_2=Q_1-S_1-S_2=6980-196-301=6483$ c.c.

$\eta m_1=.9_2$.

Settings	4/4	3/4	2/4	1/3	1/4
$\eta m_2 =$	0.86	0.89	0.89	0.86	0.75

$$(\eta m = 1 - (1 - \eta m \text{ from the table}) \times \frac{10}{P}$$

p can be calculated more accurately from formula 5.

$$p = \frac{7500 \times N_2}{Q_2 \times \eta m_2} = \frac{7500 \times 5}{6483 \times \eta m_2} = \frac{5.78}{\eta m_2}$$

This gives for the various settings :—

	4/4	3/4	2/4	1/3	1/4	
$p =$	6.72	6.5	6.5	6.72	7.7	atmospheres.

As a result of this :—

Settings ...	4/4	3/4	2/4	1/3	1/4	
$S_1 = (p+0.5)28$	202	196	196	202	230	c.c.
$S_2 = (p+0.5)43$	310	301	301	310	352	c.c.
$Q_2 = Q_1 - S_1 - S_2$	6468	6483	6483	6468	6398	c.c.

Revolutions of Motor, per minute.

Setting ...	4/4	3/4	2/4	1/3	1/4	
with $q_2 =$	2222	1666	1111	741	555	c.c.
$n_2 = \dfrac{60 Q_2}{q_2} =$	174	233	349	523	692	

The power requirements of the pump can be calculated from formula 3, and for the different motor settings are :—

	4/4	3/4	2/4	1/3	1/4	
$N_1 = \dfrac{Q \times p}{7500 \times \eta m_1} =$	6.8	6.58	6.58	6.8	7.8	h.p.

The overall efficiency amounts to :—

	4/4	3/4	2/4	1/3	1/4
$\eta = \dfrac{N_2}{N_1} = \dfrac{5}{N_1} =$	0.74	0.76	0.76	0.74	0.64

The volumetric efficiency ηv is :—

	4/4	3/4	2/4	1/3	1/4
$\eta v = \dfrac{Q_2}{Q_1} = \dfrac{Q}{6890} =$	0.927	0.93	0.93	0.927	0.852

The turning moment can be calculated by the formula :—

$Md = 71620 \times \dfrac{N_2}{n_2}$ and for the various settings are as follows :—

	4/4	3/4	2/4	1/3	1/4
$Md = \dfrac{71620 \times 5}{n_2} =$	2055	1535	1025	684	517 kg.cm.

Regulation through adjustment of the pump.

For $N_2 = 5$ h.p. and $p = 10$ atmospheres.

(Formula 5).

$$Q_2 = \frac{7500 \times N_2}{p \times \eta m_2} = \frac{7500 \times 5}{10 \times 0.91} = 4120 \text{ c.c.}$$

(Formula 4).

$$n_2 = \frac{60 \times Q}{q_2} = \frac{60 \times 4120}{222} = 111 \text{ r.p.m.}$$

With 5 h.p. the total adjustment is between the limits of 111 : 692 or 1 : 6.23.

The settings of the pump body for $n_2 = 111$ can be calculated from $Q_1 = Q_2 + S_2 + S_1 = 4120 + 296 + 192 = 4608$ c.c.

$$q_1 = \frac{60 \times Q_1}{n_1} = \frac{60 \times 4608}{1100} = 250.$$

$$\text{Setting } 2/4 \times \frac{250}{380} = \frac{1.32}{4}$$

The volumetric efficiency decreases with increase of temperature. The mechanical efficiency, however, within the limits of temperature, increases so that on balance they may vary so that the overall efficiency remains approximately constant.

Constant Delivery Pumps.

It has been stated that it is difficult to maintain high pressures with gear pumps due to the reliance on line contact. Nevertheless some remarkable results are obtained when the pump is properly designed, double helical gears and roller bearings tending to increase the efficiency.

For feed variation, a control valve is required, so that one drawback is that the maximum power is being used at all times, due to the pump forcing the surplus oil past the relief valves, so that a comparatively large pump is required, with a large oil tank or other cooling device.

The theoretical output obtainable in gallons per rev. per unit width of tooth can be calculated from the following table. The result multiplied by 80% will give the approximate actual output.

No. of Teeth	4 D.P.	5 D.P.	6 D.P.
10	0.0162	0.0105	0.0072
11	0.0181	0.0116	0.0081
12	0.020	0.0126	0.0089
13	0.0212	0.0139	0.0095
14	0.0228	0.0146	0.0101
15	0.0242	0.0155	0.0108

Example: Required delivery from gear pump running at 800 r.p.m. gears 12 T, 6 D.P. $1\frac{3}{4}$ in. face.

$$\frac{0.0089}{1} \times \frac{7}{4} \times \frac{800}{1} \times \frac{80}{100} = 9.96 \text{ galls. per min.}$$

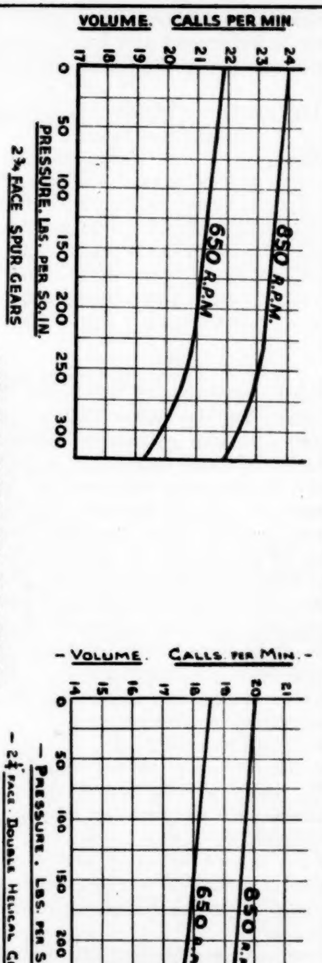
The values given are for gears with 20° pressure angle, this allowing a less number of teeth to be used than for $14\frac{1}{2}$°, besides preventing undercutting and interference. Any other values can be obtained by setting out teeth to a large scale and measuring the areas with a planimeter. Each cubic inch of space obtained is then equivalent to 1/277.3 gallons. The output is, of course, largely governed by the pressure, and two examples from actual test under severe conditions are shown.

A range of heavy duty pumps is made by Messrs. David Brown & Sons, Ltd., Huddersfield. These incorporate "Roloid" double helical rotors and represent a most successful application of tooth rotors for dealing with fluids under pressure. The design is such that they afford a higher volumetric and power efficiency than previously obtainable, together with a quieter operation due tc uniform discharge and a minimum of mechanical noise.

The special tooth form adopted affords a low-velocity rolling displacement, free from pulsation, thus ensuring a uniform oil flow. A continuous oil seal is obtained across the entire face width at every stage of engagement, with consequent high suction capacity, while the enclosure of the rotors is such that motion of the fluid takes place largely in an axial direction, hence the actual velocity of inlet or expulsion barely exceeds the peripheral velocity of the rotors.

Volumetric efficiencies of from 90 to 95% are obtained and oils of any class can be dealt with in pressures up to 300 lb. per sq. in. The pump can also be used as a motor giving a wide range of speeds with uniform torque ; there being no dead points, full torque is developed in any angular position.

The nominal discharge in gallons per minute can be seen from the table, although the actual discharge of a pump depends upon



CAPACITY CHARTS FOR GEAR PUMPS.

FIG 4.

the volumetric efficiency obtained at a given pressure and the speed of operation is influenced also by the viscosity of the fluid so that a slightly lower figure than that given may be obtained.

"Roloid" Heavy Duty Pumps. Nominal Discharge. (Galls. per min.).

Face. (inches).	R.P.M.				
	200	500	1,000	1,500	r.p.m.
1.95	1.98	4.95	9.9	14.8	galls.
2.25	3.52	17.6	35.2	52.8	,,
3	16.6	41.5	83	125	,,
4.5	24.8	62	124	186	,,
5.4	43.2	108	216	324	,,
6.75	84.6	212	423	636	,,

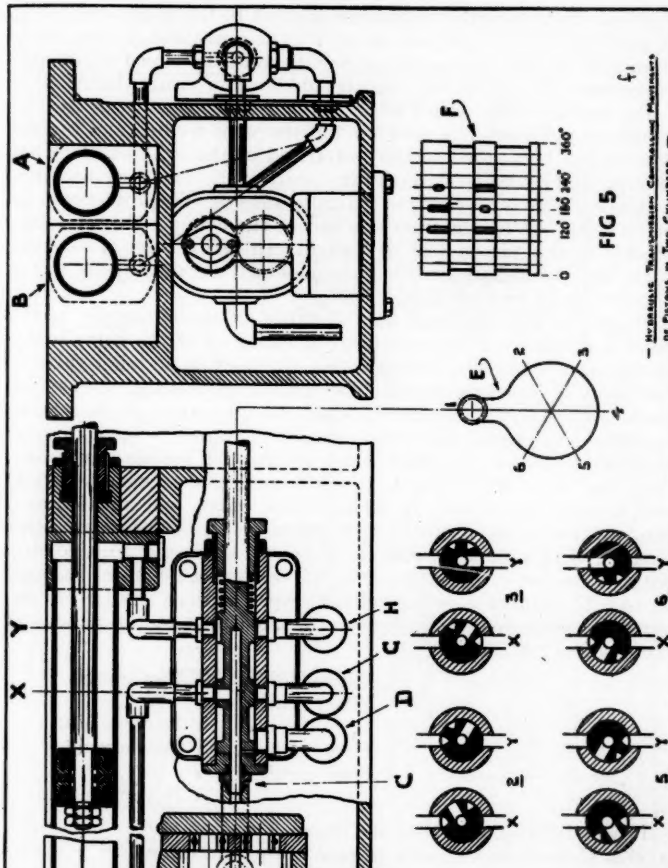
Valve Gear.

With the hydraulic operation of machines the main problems arise with valve gear, for the power and output of pump units follow definite laws and like cylinder diameters can be readily estimated. The design of valve equipment, however, is much more complex, especially if required for machine tool purposes, for it may be necessary to reverse instantly as in grinding up to a shoulder. Moreover, in most cases the speed range must be wide and operate under all condition of feed, rapid traverse and reverse, without vibration or shock which would produce inaccuracies in the work. These conditions are modified in the case of a variable delivery pump passing through a neutral position, but become extra severe with a constant delivery pump dependent upon a relief valve. In this case, oil under pressure must be rapidly cut off by valve mechanism alone, and any defect in design is liable to cause wiredrawing, violent stresses in the pipe lines, or overheating of the transmission fluid.

While there are many types of valves available, it is proposed to describe two only, firstly a rotary and secondly a piston type valve, and to combine with the first the mounting of a gear pump and cylinder arrangement.

The function of the valve is to supply and control the oil delivery to either of the duplex cylinders carrying pistons rods suitable for two separate feeding motions, as, for example, sliding and surfacing motions of a lathe.

Pressure oil enters at C, and passing into the bore of the valve is free in the position shown, to pass to the left hand end of the cylinder A. At the same time the exhaust oil coming from the opposite end is free to pass into grooves on the valve to the exhaust pipe D and the oil reservoir in the bed. A 60° movement of the control lever E mounted on, say, a lathe apron, changes the position of the valve


A
B
X
Y
H
G
D
C
E
F
0
120° 180° 240°
360°
FIG 5
— Hydraulic Transmission Controlling Movements of Pistons in Two Cylinders. —

ports, so that a reversal of traverse is given to the piston rod. This in indicated in the valve layout at F°, the positional change being from 180° to 240°, and as the oil duct is gradually closed during this movement, a variable feed can be obtained in either direction.

By rotating the control lever through part of a turn, the oil supply to cylinder A is shut off and the same feeding and reverse motion can be applied to the piston rod in the cylinder B by oil entering and returning through the pipes G and H. On many machines a hand traverse is required, so that it is necessary for provision to be made for a free circuit from one end of the cylinder to the other, otherwise the piston will be locked. In many cases a separate valve box is required for this purpose, but the arrangement shown allows this in any position.

It will be also apparent that shutting off the oil supply as the lever is rotated will cause pressure to build up in the valve unless provision is made to counteract this, for there is no relief valve in the pump or pipe line circuit. The control valve is therefore spring loaded, so that the surplus oil which, under normal feeding rates, is allowed to escape in limited quantities throuh a small duct as it enters the valve chamber, may, as pressure becomes excessive, force the valve to the right, thus uncovering the exhaust port D and allowing oil return. In addition, the spring will function as a safety valve in the event of obstruction on the slides or excessive feed, so that a table or slide may be run against a positive stop without danger of breakage. The valve section diagrams through X and Y, number 1 to 6, and the lever positions similarly numbered, give the following sequence of movement to the pistons in the cylinders A and B.

Cylinder	A,	position 1.	X Forward feed.	Y Free circuit.
,,	A,	,, 2.	X Free circuit.	Y ,, ,,
,,	B,	,, 3.	X ,, ,,	Y Reverse feed.
,,	B,	,, 4.	X Forward feed.	Y Free circuit.
,,	B,	,, 5.	X Free circuit.	Y ,, ,,
,,	A,	,, 6.	X ,, ,,	Y Reverse feed.

It will be apparent that the valve lever rotation is reversible, so that there is no necessity to pass through intermediate positions to change from forward to reverse feed, for positions 3 to 4 controls these functions for cylinder B, and positions 6 to 1 for cylinder A.

Piston Type Valves.

Before describing a piston valve arrangement, it may be pointed out that one of the difficulties of employing gear pumps on some machines is that there is a sudden jump forward when drills or cutters leave the work due to the reduced pressure in the feed cylinder. Thus one of the most satisfactory arrangements is a combination of

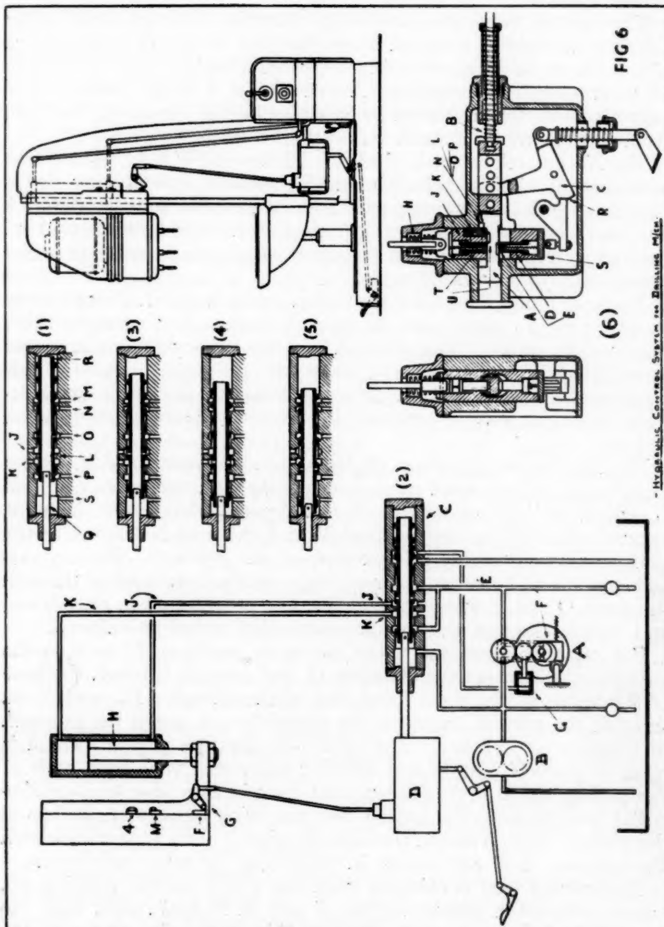


FIG 6

HYDRAULIC CONTROL SYSTEM for DRILLING M/CH.

a small capacity variable delivery pump working at high pressures for feeding purposes and a large capacity gear pump working at low pressures for rapid traverses.

The type of valve gear used may comprise a central distributing valve, controlled by stops on a machine slide by means of which two piston valves in the same casing allows the passage of oil from either or both pumps as required. Alternatively, a single piston valve may be used, this being under pressure in one direction, but held from movement by a latch on a stepped slide.

The National Automatic Tool Co., Richmond, U.S.A., incorporate a somewhat similar principle in the control system now to be described. Oil is supplied partly from a variable delivery pump A and also from the gear pump B to the piston valve C, connected to the valve control mechanism D, governed by a foot pedal, or other means to give a working cycle.

The gear pump B supplies the piston pump A and the low pressure supply pipe E, surplus oil passing through a low pressure relief valve to the supply. The pump A is of the type where an increased eccentricity of the pistons increases the oil supply, and for this purpose the lever F, which normally tends to assume a small delivery position, may be actuated by the piston G to increase the oil delivery.

With the valve in position (I), the piston is at rest, since lines from ports J and K are open to each other through the valve chamber L, which is itself closed to all other ports. Connection is made between the volume control port M and the port N leading to the tank so that no pressure is exerted on the piston G. The variable delivery pump, however, is supplying a small amount of oil through the ports O and P where it joins the low-pressure oil in chamber Q and passes through the low-pressure relief valve to exhaust.

For rapid traverse the valve moves to position (2) so that the gear pump oil enters the chamber Q, and passing through the bore of the valve to the right hand end of the chamber R, and then, through the port M, operates the piston G and causes the pump A to increase its delivery. The oil enters through port P, which is open to ports J and K and thereby connected with both ends of the piston, which moves forward at a rapid rate, due to its differential construction, the area of the rod being equal to half that of the piston. The oil forced through the pipe to port K combines with the outgoing fluid and assists in producing the rapid movement.

The cutting feed is effected when the valve reaches position (3), where connection between ports N and M is again such that the supply from the pump A to the port P is small. The oil leaves the valve chamber at the port K and enters the top end of the main cylinder. The piston H, travelling downwards, forces oil to the port J, through the holes in the valve bore and along to chamber Q,

where it combines with the oil from the gear pump and returns through the relief valve to the tank.

For the rapid traverse, position (4) is assumed. The gear pump supplies oil to port S, which passes through the chamber Q and the valve bore and enters chamber R, which is open to the port M, thus increasing the supply from the pump A. This oil enters at the port O connected to J, and passing forward returns the piston. The oil forced out of the cylinder returns by way of the port K and valve holes to the chamber Q, and again uniting with the gear pump oil returns to the tank.

The final neutral position (5) disposes of the fluid as in position (I), with the opposite sides of the piston connected through the chamber L and free of pressure.

The operating mechanism is shown at (6) and comprises a trip block " a " arranged to slide in a casing and is coupled to the valve rod. The block is normally urged to the left by the spring " b," but may be moved to the right by depressing the foot pedal which causes the cam surface of the lever " c " to engage the shoulders of the trip block, forcing it and the valve as far as possible to the right. This is the initial valve position (I). Release of the foot pedal allows the spring " b " to move the valve to the left until intercepted by the bottom plunger " d " of the latch " j " engaging the slot " e." The valve is now in position (2), and as the head moves rapidly downwards, the dog " f " is disengaged from the roller " g " and the spring " h " forces the latch " j " downwards, until it engages the shoulders of the plunger " d," disengaging it from the slot " e," while the plate " k " is lowered to engage the first step at 1 as the trip block " a " is snapped to the left by the spring " b." This movement, however, is insufficient to alter the oil flow of the valve position (2), the purpose of the plunger " d " being to prevent the trip block from movement to the left, until the head moves downwards sufficiently for the plate " k " to engage the first step at 1.

The rapid approach is changed to slow feed when the head moves downwards sufficiently for the dog " m " to engage the roller " g," thereby elevating the latch " j " and disengaging the plate " k " from the first step at " l," The valve can then move to the left by spring pressure until the plate engages the second step at " l " and position (3) is reached.

The sequence is repeated for the positions (4) and (5) by utilising the third and fourth steps at " l." Of the three dogs, " m " is the shortest, " q " of intermediate length, and " f " the longest. At any moment during the downward feed, depression of the foot pedal causes the valve to return to the position giving rapid return, or if depression is continued, to the stop position.

Problems of Hydraulic Machine Tool Operation.

In a present day survey of hydraulic machine tools, it would appear of more value to restrict comment to that type of machine where but little hydraulic development has as yet taken place. thus any remarks on grinding or broaching machines may be left for discussion or another section of the paper. It may be considered that hydraulic operation of drilling machines is well established but some of the later features may be of interest.

In this connection self-contained hydraulic drilling units are now available, being mounted either singly or built virtually around a single workpiece for high production. The simplicity of the units mitigates against the risk of breakdown while their versatility is such that they can be adapted to different products or become part of a larger or smaller machine as required.

The theory of unit development can perhaps be best exemplified by the Ingersoll Power Pack machines, which are however not themselves hydraulic. The theory is based upon seven elements in machine tool operation as follows :—

Unit	Function
(1) Fixture pedestal	To raise the work to a convenient working height.
(2) Bed wing	To raise the machine to the same convenient working height.
(3) Slide	To allow the cutting tools to approach the work and retreat from it.
(4) Saddle	To provide a means of carrying the power unit on the slide.
(5) Power unit	To rotate the tools and feed them in and out of the work.
(6) Work unit	To carry the tools.
(7) Fixture	To hold the work.

With many advantages hydraulic units can be adapted to provide the power, these being adjustable for length of stroke, rapid traverse and feed, or if required, a jump feed for machining through intermittent bores, or again, a time delay for accurate depth can be incorporated.

With the Ex-Cell-O hydraulic unit, the feed range is from 1 to 25 in. and the rapid traverse from 330 to 250 in. per minute depending on the size of the unit.

For application to standard types of not only drilling machines, but practically all other types of machines, the Oilgear Co. build complete fluid units, comprising a high pressure variable displacement pump, compensated for pressure and temperature change, a large volume gear pump for rapid traverses and all necessary valves and controls in one casing ready for coupling for dog or lever

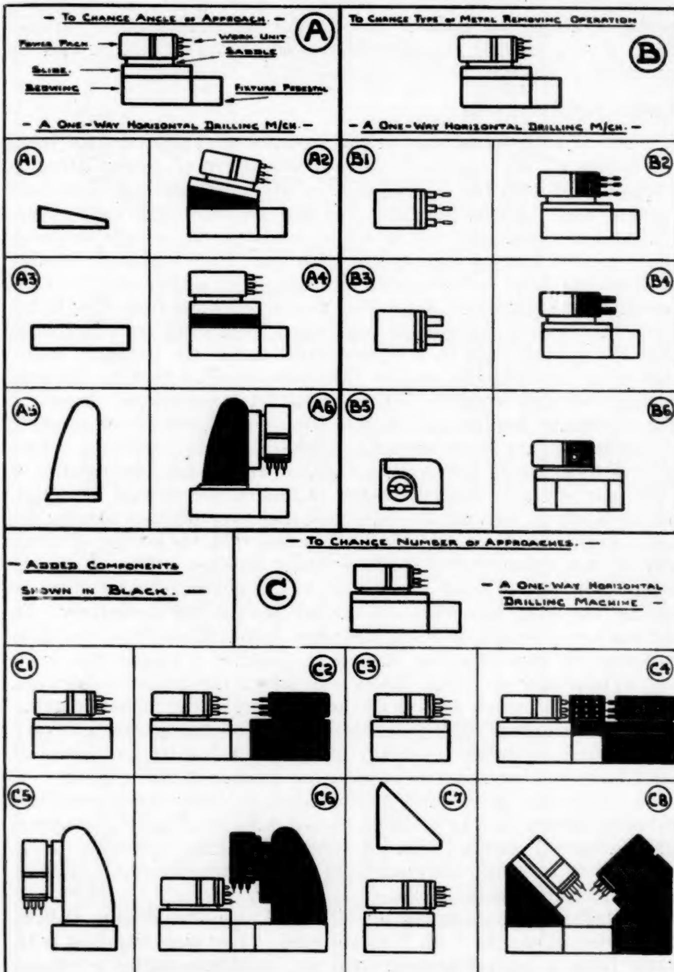

— To Change Angle of Approach —
A
Work Unit
Saddle
Fixture Pedestal
— A One-Way Horizontal Drilling M/ch. —
To Change Type of Metal Removing Operation
B
— A One-Way Horizontal Drilling M/ch. —
A1
A2
B1
B2
A3
A4
B3
B4
A5
A6
B5
B6
— Added Components Shown in Black. —
— To Change Number of Approaches. —
C
— A One-Way Horizontal Drilling Machine —
C1
C2
C3
C4
C5
C6
C7
C8

Fig. 7.

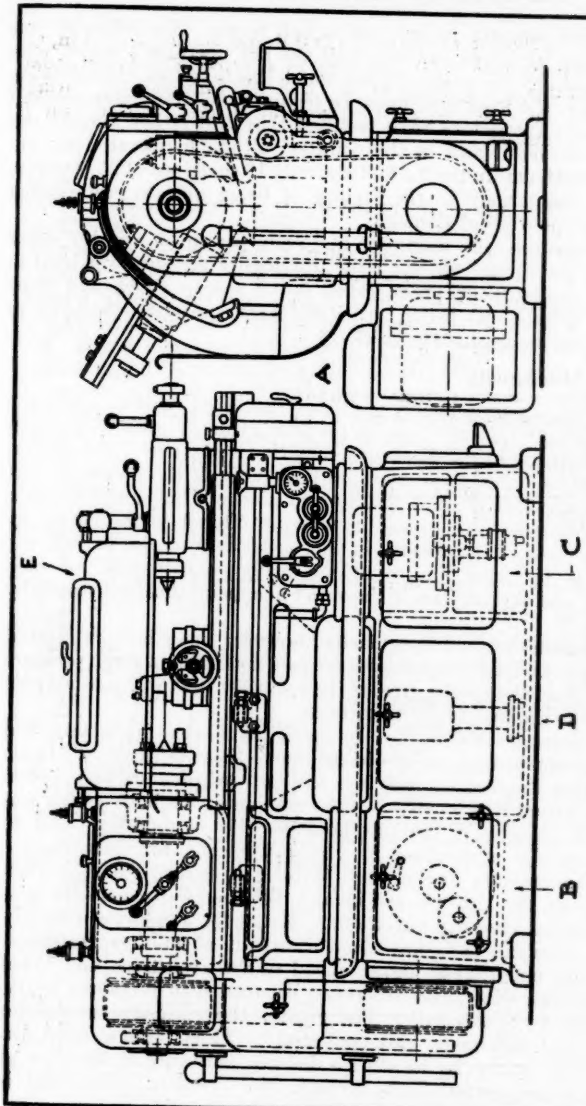
operation. A feeding pressure of 1,000 lb. per sq. in maximum, and a gear pump pressure of 50 lb. per sq. in. is provided. The standard casing contains six gallons of oil, and all functions of variable feed, dwell, neutral, rapid traverse and reverse motion can be obtained as desired.

Lathe Application.

There is little advantage in the incorporation of hydraulic feeds on lathes of the standard type, particularly if screwcutting is required, for with the development of the magazine gear box both feeding and screwcutting are provided for with one mechanism. On the other hand, for lathes of the short traverse, single or multi-tool type working at high speeds with light cuts, hydraulic feeding motions are most satisfactory. Prominent examples are the Holbrook Electro-hydraulic lathe and the Magdeburg Chip-flow lathe. Both machines make radical departures from standard practice in that the spindle runs in a reverse direction to the ordinary lathe, this being necessary to protect the operator by deflecting the long curling cuttings from the cutting zone in a downward direction. The hydraulic equipment on the Holbrook machine comprises a constant pressure pump working at 100 lb. per sq. in. giving a feed rate of 0 to 20 in. per minute cutting, and a constant return of 25 ft. per minute. Spindle speeds of 654 ft. per minute on a 1 in. bar or 3,925 ft. per minute on a 6 in diameter bar are obtainable, and accurate work length is assured by the valuable feature of being able to run against dead stops without damage. The oil simply escapes through a relief valve, and as the saddle is under pressure against the stop, shoulders are turned cleanly and accurately. In the matter of length of work, the most favourable is that less than 10 times its own diameter in length, because if longer, the work has a tendency to bend under cut when unsupported between centres, and steadies are out of the question at the highest cutting speeds. When using multiple tools on shafts requiring removal every few minutes, hydraulic operation of the tailstock by foot pedal or hand lever is advantageous in reducing fatigue of the operator.

To emphasise the accuracy obtainable, to show the possibilities of heavy cutting and to place on record one of the most courageous and successful introduction of hydraulic lathe operation, it is proposed to revert to the year 1916 and describe the machining of 18-pounder and 9.2 in. shells.

The lathes were designed by Mr. W. Littlejohn Phillip, O.B.E. and varied in weight from 1 to 15 tons. They were supplied with water from a central station with the hydraulic mains overhead and vertical pipes to each machine so that no air entered the circuits. The factory produced an 18-pounder shell from the black bar in fifteen and a quarter minutes, and a 9.2 in. shell with screwed-



— ELECTRO - HYDRAULIC AUTO - TURNING LATHE —

— MESSRS HOLBROOK MACHINE TOOL Cº LTD. LONDON. —

Fig. 8.

in base and adapter in two and a half hours. No difficulty was experienced by heavy cutting, and the net operator's wages were actually paid by the value of scrap which was taken off, thus showing how quickly the work was done. In regard to accuracy, independent Government Inspectors have testiffed that the lathes turned out some of the best work in the country, although operated by unskilled girl labour. In the 9.2 factory, only 19 shells out of 25,000 were rejected, due largely to the facility of being able to run against positive stops without damage.

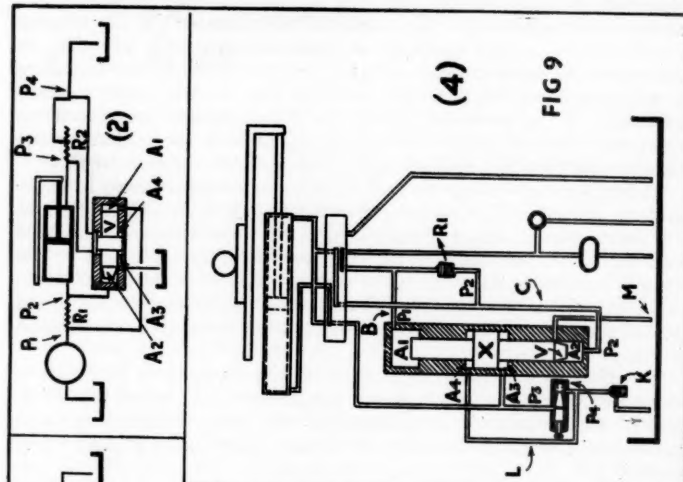
Two pressures were used, one of 750 lb. per sq. in and the other of 300 lb. per sq. in. On one lathe, the higher pressure was used in the loose headstock to press a shell into a special chuck and while its outer surface was being turned the lower pressure was used to keep the centre against the work.

Milling Machines.

The special requirements of milling machines renders hydraulic operation more difficult than in the case of many machines, particularly if cutter rotation in reverse direction is required. The general solution is to maintain pressure on both sides of the piston so that the table is locked when in the stopped position, but under traverse when any increase or decrease in pressure offered to the cutter by the work causes the pressure in the outlet line to rise or fall, the pressure in the inlet line is automatically compensated for such variation, and the pressure in the outlet line maintained substantially constant.

The question of feed irregularity, however, may be complicated by changes of oil viscosity, due to the fact that under a given pressure the more fluid the oil the greater the quantity that will pass a given orifice, thus the feed tends to increase.

The Cincinnati Milling Co. by patented devices have provided a most comprehensive means to overcome these difficulties. In an open system the feed throttle is in the forward pressure line, and in a closed system it is located in the outlet line. In both systems as the hand throttle is closing the feed rate is reducing, and to compensate for a decrease in viscosity means are provided to relieve the pressure at the hand throttle to ensure that the main flow through the throttle will be at a volumetrically constant rate irrespective of viscosity variations. Means for measuring viscosity depends upon the fact that when a stream encounters resistance its pressure potential is higher ahead of the resistance and lower after leaving. Now, the difference between such pressures is also a function of the viscosity. The greater the viscosity the greater the potential difference, and conversely. Thus, with a constant-displacement pump, by locating a definite resistance in the pipe so that the entire output flows through, the pressure differential is


(1)
(2)
(3)
(4)
FIG 9

an index of the viscosity. This is utilised to compensate for all variations in viscosity that would render the feed rate irregular.

But when the hand throttle is set at a pre-determined rate, the machine must operate not only against viscosity variation, but against the resistance of the cutter. For this latter the oil pressure must increase as the mechanical opposition increases, but only to the extent to cause a volumetrically uniform flow. Hence to ensure a constant flow, the pressure ahead of the throttle must increase as much as the pressure behind it. For complete compensation, then, the viscosity differential must operate in opposition to the throttle differential, mutually regulating the resistance to flow.

Referring to diagrams (1) and (2), R1 is a resistance fixed in value and R2 an adjustable resistance set at any definite value. P1 and P2 are pressures ahead of and behind the first resistance, and P3 and P4 ahead of and behind the second. Thus, P1—P2 is the viscosity differential, varying as the viscosity irrespective of duty, while P3—P4 is the throttle differential, which must be maintained in definite ratio to the viscosity differential irrespective of variation in duty.

To effect this, a valve V sensitive to these pressures P1 to P4 is employed, and arranged to be in equilibrium only when the effect of P1—P2 is equal to and balanced by P3—P4. Interpreting these pressures in terms of areas, i.e., as forces, then P3A3—P2A2=P3A3 —P4A4.

When dealing only with the maintenance of this ratio, regardless of changes in the actual pressure valves, the equation may be simplified by considering A3=A4 and A1=A2, but only where compensating for viscosity alone.

The equation thus becomes

$$A1\ (P1—P2)=A3\ (P3—P4) \text{ and so}$$

$$\frac{A2}{A4}=\frac{A1}{A3}=\frac{P3—P4}{P1—P2}$$

In diagram (1) if the work resistance increases, the piston slows down and pressure P4 tends to rise on account of the flow through R2, thus tending to reduce the throttle differential P3—P4. This unbalances the valve V and increases the effect of the viscosity differential P1—P2, which in turn tends to close the valve and increase the resistance to drainage. Thereupon, P2, P3, and P1 rise by an equal increment until the throttle differential is restored to its original value and maintains the same piston feed as before.

In diagram (2) the piston is ahead of the throttle, so that an increase in work resistance tends to slow down the piston, so reducing P3 and increasing P2, and by an equal increment, increasing the pressure P1. The new values P1 and P2 will be balanced without

any additional effect on the valve V. The pressure P3 being reduced enables the valve V to move in the direction of closure and increase the resistance to drainage, so that the increased piston pressure maintains the feed without variation. The arrangement also compensates for viscosity change as in diagram (1).

To obtain the differential of higher and lower pressures, it is necessary to apply them to appropriate areas arranged to exert opposing forces on a valve-actuating member. In diagram (4) a plunger X is arranged to slide in a cylinder divided into four chambers, and presents four distinct areas, A1 and A2 being in opposition, and likewise A3 and A4. A pipe at B leads to one chamber and subjects the area A1 to the pressure P1, so that the piston is given a force P1A1. Similarly, a pipe C subjects the piston to an opposing force P2A2, the resultant being directly proportional to the viscosity differential and its direction determined by the algebraic difference of the two products, which is independent of the actual size of the two areas. In a similar manner the throttle differential may be reduced to a balancing pressure on the same valve.

Referring to diagram (3), the four areas may be reduced to three, as shown in full lines, as against the separated dotted portion A2. In operation, the constant-delivery pump D supplies oil to the control valve E, which, for rapid traverse, is passed direct to the feed cylinder, the discharge being through the pipe F to the tank. For feeding purposes the control valve is set to connect the pipe G with pipe H, which leads to the viscosity compensator, and then through the pipe J the oil passes to the control valve for the feed. The viscosity determiner comprises a resistance R1 located to pass the full pump output, the oil dividing immediately after, some going forward to the control valve and some to exhaust through the compensator at a volumetrically constant rate, regardless of viscosity, but determined by the setting of the throttle R2 for the actual rate.

This throttle may be located ahead of or behind the feed piston, manually adjusted or automatic, as indicated in the various diagrams.

In diagram (3), where P2 and P3 are equal, it is sufficient to use a single chamber by making the end of the plunger equal in area to A2 plus A3. In diagram (4), P4 is fixed by the magnitude of the relief valve K, which if omitted, enables the line L to be eliminated. In each instance, a tapered portion V of the valve member X serves to restrict the size of a side outlet, thus constituting a valve for regulation of the exhaust through the pipe M. The area A1 must equal A2, and A3 must equal A4 to conform to the general relation.

$$\frac{\text{Area submitted to throttle differential}}{\text{Area submitted to viscosity differential}} = \frac{\text{Viscosity differential}}{\text{Throttle differential}}$$

If the plunger areas A1 and A2 be represented by Av and the annular areas A3, A4 by At, then the condition of equilibrium becomes

$$\frac{At}{Av} = \frac{P1 - P2}{P3 - P4}$$

Reciprocating Machine Tools.

The real problem of hydraulic operation of machines commences with planing machines, although again there are examples from the past indicating that hydraulic operation is no new thing. Messrs. Tannett, Walker & Co., of Leeds, were using six heavy planers hydraulically driven in the year 1880, and although slow running were stated to be capable of heavy cutting and a big improvement over the noisy geared machines of that day.

To-day, however, the demand for high speeds, coupled with the efficient electrical drives available, has made the planing machine the least suitable of all machine tools for hydraulic operation.

The attempt to introduce hydraulic transmission is based upon a desire to eliminate the momentum which arises from the driving members, this being about 30 times that of the table and work, and it is claimed that if this is reduced, quiet running and work free from chatter marks should result.

The chief difficulty is created by the high table speeds required and the enormous stresses set up at the point of reversal. The difficulties are also increased by the length of the cylinder, and many ingenious designs have been evolved to overcome the problem. It is preferable that the piston rod should be subjected to tensile stresses only, thus duplex cylinders are required operating in a central cylinder block. Another device takes advantage of the fact that with reciprocating machine tool drives, the maximum power and maximum speed are not required simultaneously. Two cylinders A and B differing in diameter, are employed so that oil can be supplied to either cylinder, or simultaneously to both if required. Oil supplied to the large cylinder A, produces a low speed at high power for roughing, and when supplied to B gives a high speed with lower power for finishing cuts. The same quantity of oil supplied simultaneously to both cylinders produces a still lower speed.

An arrangement by Waldrich utilises a piston rod fixed in the bed and bored hollow from both ends close up to the piston in the centre. Discharge slots at either side of the piston in turn allow the exit of oil from the hollow rods to the cylinder block fastened to the underside of the table, causing the table to traverse at a speed determined by the amount pumped into the rods. Another device by the same firm utilises a double cylinder block, sliding along the machine bed and having two cylinder bores communicating at

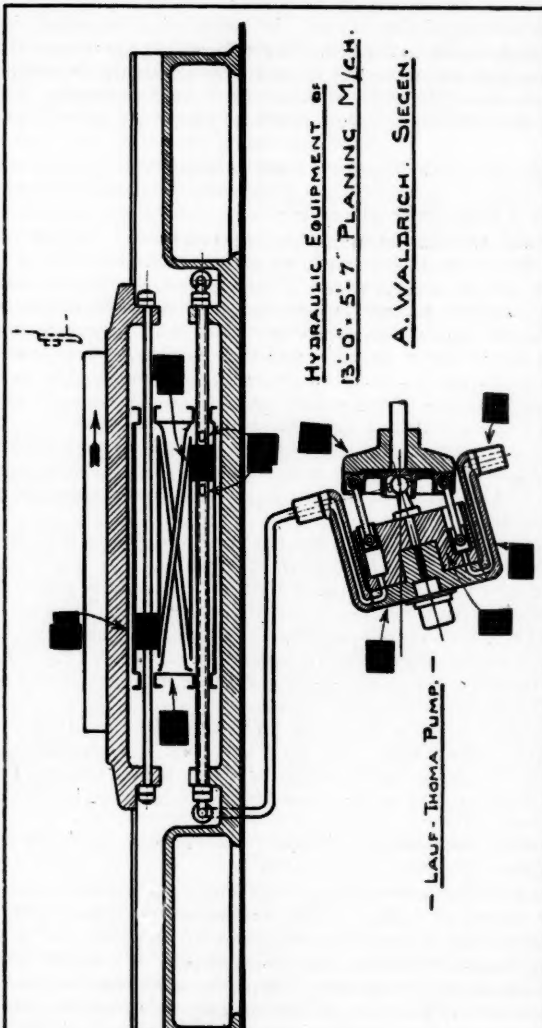

Hydraulic Equipment of
13'-0" × 5'-7" Planing M/ch.
A. Waldrich. Siegen.
— Lauf-Thoma Pump. —

Fig. 10.

opposite ends by similar ducts. As before oil is supplied through tubular piston rods to the first cylinder bore, but now passes through the connecting ducts to the second cylinder to traverse the table piston. As the cylinder block, however, is itself traversing, the table now moves with equal force, but with twice the speed along the bed.

In spite of much ingenuity, particularly by German designers, the difficulty of overcoming a jerky traverse and a sluggish reversal of a long stroke planer is no easy matter, but more success is apparent with machines of the planer-shaper or ordinary shaper type. With the latter machines many advantageous hydraulic features can be incorporated. These include cutting speeds and return independent of each other, the power is applied in a straight line without any lifting tendency of the ram, there are few fast moving parts subject to wear, and all advantages of stepless speed are available. At the present time, no British machine is marketed but considering the design submitted, a hollow draw rod is fitted to provide a quick return, which, differing from the mechanical drive, remains at a constant ratio for any length of stroke. A subdivision of ram areas on the pressure side enables a high finishing speed to be obtained in addition to a high powered lower speed range for roughing cuts.

The pump draws oil from the reservoir and passes it to the control box mounted at the operating position. The box comprises a series of rotary valves operated by four levers giving " stop and start " " hand reverse," "speed control " and " change-over " respectively. The automatic reverse motion is actuated by adjustable dogs on the ram giving a maximum stroke of 12 in. Part of the oil from the pump passes to a valve at the rear of the control box. This valve is actuated simultaneously with the reverse valve and supplies oil to a piston in the feed box immediately below the main control box. The feed is actuated hydraulically, but varied mechanically by a feed lever which masks a ratchet wheel. All the levers on the machine are fitted with Murray Colour Control.

Hydraulic Couplings, Torque Converters and Hydraulic Drives.

The use of hydraulic couplings and torque converters has increased rapidly in recent years. The fundamental difference between the two is that the former comprises an impeller and a runner working together without the interposition of a guide wheel or reactionary member, whereas a torque converter requires stationary guide vanes to deflect the oil flow usually as a means of obtaining a higher torque at a correspondingly lower speed at the output shaft. In both cases the torque transmitted is dependent upon the velocity of movement of the working fluid.

Of hydraulic couplings, the most familiar is the fluid fly-wheel of the two element type, incapable of torque conversion, so that a gear box must be used to augment the engine torque for rapid acceleration and hill climbing, although it is practicable to drive a car in top gear under all traffic conditions.

Unfortunately, a compromise must be made between two opposing factors to obtain the most efficient size of fluid coupling. On one hand, to obtain a minimum of power loss and heating, the slip must be small for average running, indicating the requirements of a large profile diameter for the oil circuit. But when the engine is idling with the gear engaged, there is a tendency to creep if the oil slip is not high or the drag torque low. This suggests a coupling of small diameter, so that the problem must be solved by a satisfactory compromise between the two.

These difficulties can be overcome by the incorporation of a traction coupling providing a reservoir chamber on the back of the driven part, the object being to remove part of the oil from the working circuit under starting conditions, when the slip is between 50 and 100%, so as to reduce the creeping tendency and assist the engine to pick up load, and secondly to return the oil to the circuit when a certain speed is reached.

A further refinement is an anti-drag baffle comprising a circular plate fixed near the inner profile diameter of the driving or driven member, this baffle again reduces the creeping tendency when idling by restricting the oil flow, but it has no effect when the normal speed is reached. The cushioning effect on a fluid fly-wheel with these two refinements is at least three times as great as when not fitted, thus safeguarding the transmission against shock. In addition, the difficulty of gear changing due to the drag problem has disappeared by combining the fluid fly-wheel with the Wilson type epicyclic gear box, where friction bands take the place of sliding gears.

In regard to torque converters, a general characteristic is a very high efficiency at a certain speed ratio, but at other speeds the efficiency falls rather fast. With certain types, however, such as the the Lysholm-Smith torque converter, it has been possible to obtain a flat efficiency curve, averaging from 85 to 89% by means of improved turbine blading, which enables the torque speed range to become automatically variable within a wide range of high efficiency.

Extensive experimental work with torque converters has been successfully carried out by Leyland Motors, Ltd. With one arrangement in order to utilise the advantages of the converter in regard to shockless torque, along with the automatic variation in the torque speed characteristic, without having to obtain the higher speeds on a falling efficiency curve, a direct drive is introduced

by means of a two-way clutch which disconnects the pump from the engine and connects the fly-wheel directly to the output flange of the converter. The turbine, which drives the output shaft through the medium of a fly-wheel, automatically disconnects itself from the drive and the whole hydraulic system comes to rest.

The complete unit which is bolted direct to the rear of the crank-case, comprises in order from the engine end, a double acting clutch as described, the hydraulic system, the freewheel and reverse gear. The fluid used is 95% paraffin oil plus 5% of lubicating oil.

On starting a vehicle the torque converter clutch is engaged at idling engine speed and the torque is built up by increasing the engine speed on the converter and shockless motion ensues when the torque, which is approximately proportional to the square of the speed, attains a sufficient value. When the vehicle reaches the required speed, say 20 miles an hour, the toggle mechanism is moved over and the direct drive engaged.

The clutch control may be moved back to the torque converter at any time without shock. When driving in traffic the driver has complete freedom for steering, control being entirely by the accelerator pedal, the torque speed changes being entirely automatic, a valuable safety feature.

To obtain the best results from a torque converter it must be used with an engine of suitable characteristics, in similar manner to a ship's propeller designed to suit a given engine speed. The torque speed characteristic can be altered if required, either by change of the pump wheel diameter, change of blade angle or by incorporating reduction or speed increasing gearing.

It must not be assumed from the two examples mentioned that the use of hydraulic couplings and torque converters is restricted to automatic vehicles, for both transmissions have wide application in industrial drives, including chain grate stokers, laundry calenders, coal pulverisers, hydraulic presses for plastic moulding, cranes and winches, in fact one coupling made by J. M. Voith Maschinenfabrik for centrifugal pumps, transmits no less than 36,000 h.p.

While the automatic variation of torque with speed is useful with the torque converter, this feature whereby speed is responsive to load is unsuitable for machine tool drives where the transmission depends upon the transfer of fluid under high pressure and at a relatively low velocity. There is a wide field of application in which either the turbo-type or the displacement type of transmission can be employed, the efficiency of the first being superior at the high speed end of the range, but falls well below that of the displacement type over a wide speed range as required for machine tools.

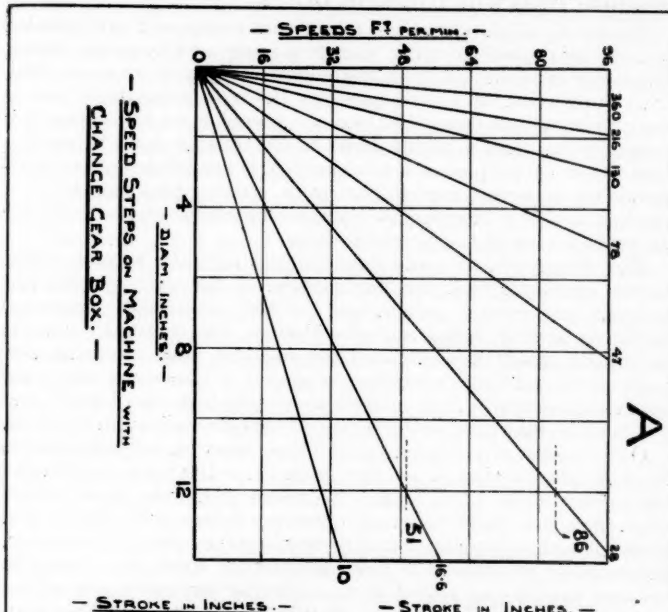

A
— SPEEDS FT. PER MIN. —
0
16
32
48
64
80
96
360
216
130
78
47
28
86
51
16·6
10
4
8
12
— DIAM. INCHES. —
— SPEED STEPS ON MACHINE WITH CHANGE GEAR BOX. —
B
— STROKE IN INCHES. —
2
4
6
8
10
12
16
20
24
300
50
30
70
36
20
11
CUTTING SPEED. FT. PER MIN.
— CAPACITY CHART FOR SHAPING M/CH. WITH CHANGE GEAR BOX. —
C
— STROKE IN INCHES. —
90
10
10 STROKES PER MIN.
CUTTING SPEED FT. PER MIN.
— CAPACITY CHART FOR SHAPING M/CH. WITH HYDRAULIC MOTOR DRIVE. —

Fig. 11.

Machine Tools with Hydraulic Drives.

Practically all the piston and rotary vane pumps can be duplicated to form two combined units, namely a pump and hydraulic motor, the latter receiving oil from the pump to produce rotary motion. This duplication, of course, increases the cost of the drive, and as the efficiency is the combined one of a pumping set and motor, it is generally less than a geared drive, nevertheless it must be remembered that the purpose of a machine tool is not efficiency as understood for a prime mover, but work output plus accuracy of production. For this purpose then, strong claims can be made for an extended use of the hydraulic drive.

The elimination of many opportunities for D.C. variable speed motor drives through the introduction of the grid system, has brought into greater prominence the lack of suitable means for infinitely variable speed control. Further, the continued increase in cutting speeds has extended the required range tremendously particularly on those machines requiring a low speed range for machining specially hard materials or using high speed tools, and yet must be capable of using cemented-carbide tools when required.

One solution is a duplex speed range, that on a German lathe for example, varying in one case from 15 to 190 r.p.m. and in the other from 95 to 1,180 r.p.m. To cover such wide speed ranges often means a duplication of operating levers with instruction plates of such a complication as to bewilder the operator, moreover, the inability to obtain the most satisfactory speed must often be counted against any geared drive. Consider two cases, one a gear box with eight speeds, 10, 16, 7, 28, 47, 78, 130, 216, and 300 r.p.m. The difference between the speed ranges is 67%, so that assuming a shaft 12 in. diameter is to be machined at 65 ft. per minute, either the speed 16.7 or 28 must be used, giving cutting speeds of 51 or 86 ft. per minute. As the tool would not stand the higher speed of 30% more, the lower speed must be used with a loss of 25%. In a similar manner the illustration of the shaping machine charts, indicates that if a speed of 40 ft per minute with a 10 in. stroke is selected as suitable for a given material, the operator is restricted to either 30 ft. or 50 ft. with the drawbacks as before, whereas with the hydraulic drive the exact speed may be used under all conditions.

Surveying a few hydraulic drives, on one type of the Magdeburg lathe headstock, a Lauf-Thoma drive is fitted, giving a speed range of 60 to 1 this being supplemented by a double back gear. The hydraulic gear is provided with instantaneous reverse and safety valves, and when the machine is stopped the spindle is hydraulically braked. The change of spindle speed is effected by handwheel on the headstock, or if required, from a central control station, and the superiority of the drive is still further enhanced by the fact that

speed variation can be made whilst cutting. Thus while using carbide tools the cut can be started at a speed which will not damage the tool, and then with the tool in full cut the speed may be set to the maximum. When desired the surfacing feed can be connected to the control handwheel so that the speed is automatically increased for small diameters or decreased for large.

Tests have been carried out to ascertain the comparative efficiency and power consumption of one of these lathes and a similar machine fitted with an all geared headstock.

The lathes were of 10 in. centres, with a speed range of 9.4 to 600 r.p.m. in the case of the fluid drive and 9.4 to 480 r.p.m. in the other case. The power consumption was measured by electric meters, while the work output was ascertained by a brake resistance, and by actual work with a tool on a test piece. In the case of the fluid drive at normal load of 9.6 h.p. and 175 r.p.m., the efficiency proved to be 78% with overload this was increased to 82%. The gear driven machine gave an efficiency of 85% at normal load and 200 r.p.m. this rising at overload to 86%. At speeds ranging from 300 to 480 r.p.m. the efficiency of the machines was about equal but of a lower value ; at 480 r.p.m. the figure was 59%.

In the mechanical tests, standard cuts were compared on the two lathes, and the actual cutting pressure at the tool was measured. The average cut was of 0.0093 to 0.0108 sq. in. cross section are with a cutting speed of 1,256 ft. per min. In general the two machines showed but little difference, but with equal power consumption and at the same speed, the geared machine gave a 9% better output. Taking into account the cost of power and the overhead working costs of the lathes per hour, the following conclusions may be drawn : Although the hydraulic lathe yields only a 10% higher ouput of work with 21% higher consumption of power when working at full load, it actually affords a saving of 7%. At half load, the oil drive gives a 13% higher output with a 31% higher power consumption, which taking into account the cost of power and lathe time, represents a saving of 10%.

Where the hydraulic drive is required for light feeding duty and not for a main drive, a simple type of pump and motor unit can be used without the cost and complication usually entailed. On a range of grinders, the Landis Tool Co. use a low-pressure variable flow pump on the rotor principle. The average pressure is 70 lb. per sq. in. and a departure is made from the conventional piston and cylinder arrangement for the table traverse. The oil is drawn from a tank in the bed by the variable flow pump and passed to a hydraulic motor of simple construction of the two chamber type with two rotors keyed to the same shaft, but with eccentrics set 180° apart. From the motor shaft the transmission is by the usual mechanical elements, worm and wheel, pinion and rack. The table

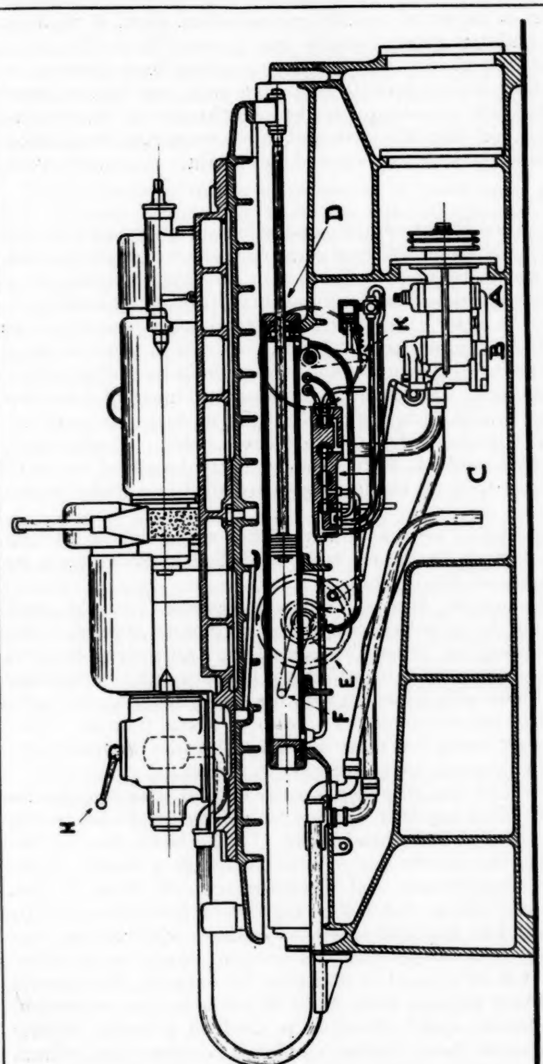
traverse can be varied from 12 in. to 20 ft. per minute by a small handwheel which regulates the amount of oil pumped and in turn regulates the motor speed.

It is a peculiarity of machine tool design, that while great numbers of machines are made with oil controlled feeds and a lesser number with hydraulic drives, there are comparatively few which combine the two. These are restricted to a very few grinding machines, shaper-planers and honing machines. Remarkable as it may seem, the only British representative would appear to be the Precimax cylindrical grinding machine shown. The hydraulic equipment comprises two Enor type pumps located in the oil reservoir, one supplying oil through the reversing valve to the cylinder for the table traverse, variable from 6 in, to 240 in. per minute without steps. For plunge cut grinding the table travel can be reduced to only 0.2 in. The handwheel for the table traverse is automatically disengaged by oil pressure when the table power traverse is engaged and is similarly engaged when the power traverse is stopped.

A built-in hydraulic motor drives the work head, the oil being supplied from the second pump. The work speeds can be regulated without steps from 0 to 450 r.p.m., the hydraulic motor giving an increased torque with reduced speed; i.e., constant horse power.

Two methods of feeding are available, one continuous and variable for straight-in plunge cutting and the other for intermittent feeding when traverse grinding. In order to minimise time in work changing, the wheel head is withdrawn rapidly by oil pressure a distance approximately 2 in., and after inserting another piece, the head moves forward rapidly by the same means. The tailstock is operated hydraulically by a foot pedal, leaving the operator with both hands free to handle long or heavy work, but as a safeguard, the pedal cannot be operated with any part of the machine in motion. This would appear to comprise the most comprehensive hydraulic equipment found on any machine tool, comprising as it does, no less than sixteen distinct functions all controlled from the operator's standing position.

The Mayer and Schmidt honing machine affords another example of the combined feed and drive by oil pressure. A characteristic feature of honing machines, differing from that of any other machine tool, is the combined rotary and reciprocating motion of the spindle. This does not introduce any pronounced difficulties in the way of hydraulic transmission for reciprocating the spindle, although mechanical means are as yet mostly adopted for the rotary motion. The machine in question, however, utilises hydraulic means for both motions with advantage. These comprise two Enor pumps at the base of the machine, one supplying oil to a vertical cylinder for the spindle reciprocation, and the second driving a hydraulic motor at the top of the column. The eccentricity of both pumps can be



— Vertical Section Showing Hydraulic Equipment for Table Traverse & Work Head Drive. —

— 'Precimax' Cylindrical Grinding Machine. John Lund Ltd. Keighley. —

Fig. 12.

varied, giving a range of spindle reciprocation from 0 to 50 ft. per minute, and the rotary spindle speed from 75 to 270 r.p.m. When honing commences, the levers controlling these motions are coupled together, and so operate the valve gear that honing starts slowly and gradually speeds up, also by reducing the oil supply just before reversal, the traverse is slowed down to reduce shock liable to cause the hones to suddenly expand and produce inaccurate work.

Conclusions.

It is not claimed that hydraulic operation is the panacea for the many difficulties of speed and feed control on every type of machine, nevertheless the increasing application in all directions is a proof that it can be applied to practically all machines, small or large.

In regard to the latter, this is well understood in the case of presses but machine tool examples are not lacking. A case in point is a large drilling machine for steel columns and girders, built by the Structural Service Co., Chicago. The machine comprises two horizontal and one vertical drilling saddle, each carrying 32 drills, feeding by a hydraulic cylinder giving a pressure of 40,000 lb. to each saddle. The bedplate for the horizontal saddle is 47 ft. long, while the length of the table is 110 ft. At one traverse the 96 holes are drilled in a girder length of 2 ft.

With a crew of five men, a column 24 ft. long, weighing $2\frac{1}{2}$ tons and having cover plates $12\frac{1}{2}$ in. wide, is drilled in 25 minutes by twelve operations of the machine.

As another example, Messrs. Churchill Machine Tool Co. have built a machine for grinding operations on heavy tapered roller bearings, this being for British Timken, Ltd., and weighs 45 tons.

It is thus to hydraulic drives that we look for further developments, for mechanically infinitely variable speed devices, although, vast in number, are restricted to a ratio of about 9 to 1. Pole-change motors are useful but more limited still, while the incorporation of a Ward-Leonard set is not always warranted.

Yet another speed changing idea is a mechanical change hydraulically operated. This has been applied to drilling machines, boring and turning mills and lathe headstocks. In the latter case on the Lehmann lathe the speeds are selected through a rotary valve controlling the engagement and disengagement of three friction clutches and three sliding clutches giving sixteen forward and eight reverse speeds. The required speed is obtained without stopping the machine or even disengaging the friction clutch, so sensitive is the control that at a speed of 96 r.p.m. for example, the spindle can be started and stopped from 20 to 30 times in one revolution. This same hydraulic speed changing is used on a radial drilling machine, oil pressure being further employed to clamp the column to the base plate when drilling and to lift it slightly for rotation.

Similarly, hydraulic power is employed for raising and lowering the arm and clamping it to the column.

It can be demonstrated then, that hydraulic operation has invaded every section of the mechanical world, not only for transmission purposes but for other uses such as hydraulic loading of spindle bearings as on the Churchill grinders, variable pitch aeroplane propellers controlled by hydraulic valves, ship's steering gear and gun controls.

The adoption of oil for transmission is based upon many advantageous grounds including, anti-corrosive and lubricating properties, chemical reliability of mineral oils and practical incompressibility which prevents the generation of heat and loss of energy. Due to the bulk modulus of compressibility, there is a slight cushioning effect, but leakage past the valves, etc., is more responsible for cushioning than oil elasticity. It is this feature that makes hydraulic operation unsuitable for cases where an accurate relation must be maintained between two co-ordinated motions.

The occasional difficulty of sluggishness at starting is easily overcome by the selection of a suitable oil which should have a good cold or pour test. The "Arctic" grades of oil fulfill the required conditions with a pour test of —30°F., so that free flowing obtains at freezing point. The specific gravity is 0.902 and the viscosity in Redwood seconds, varies from 418 at 70°F., 160 at 100°F., 72 at 140°F., 47 at 180°F., to 42 at 200°F. Generally speaking the lightest bodied oil that can be used will give the best results, providing there is no undue leakage, for an increase in viscosity increases the critical velocity, this being the velocity of oil at which the flow changes from viscous to turbulent. The speed of water in hydraulic installations is always above the critical range but for oil transmission a steady flow is required, 8 ft. per second being a suitable basis on which to work.

(NOTE—*This paper by Mr. Town was awarded the Medal for the best paper by a member during the* 1936-37 *session*).

Discussion—Leicester and District Section.

MR. J. H. BINGHAM (Section President, who presided) : Personally, I must confess that my knowledge of hydraulically operated machines apart from presses of which I have had some slight experience, is limited. My experience of hydraulic operation and control to machine tools is more limited still. I look to some others in the audience therefore, to put some questions relating to machine tool actuation. During the lecture, however, there were one or two questions which were looming up in my mind. One of them was in connection with " cushioning " to which Mr. Town referred almost at the close. I recollect some years ago hearing, or reading, a discussion, or reading a paper on a similar subject, and I recollect that " cushioning " was one of the limitations, or limiting factors, to the definition of hydraulic control to machine tools. Mr. Town explained that it was impossible to definitely limit the travel when hydraulically controlled, owing to the compressibility of oil, distinct from the usual general opinion that oil, or water, are incompressible. There is a certain degree of compression. I cannot give it in figures, nevertheless it is there.

I also recollect that another drawback was stated, and that was of " jumpy travel." By jumpy, I mean distinct from the chattering that Mr. Town referred to. Think again of milling machines for instance—if depth of cut is varied, due to irregularity in form of material, it is quite understandable that there will be cushioning, or variation in the motion. I am told that that jumpy motion is in no way detrimental, and that the results are just as good as if the motion were uniform. One important omission to my mind in the lecture, is to deduce the advantages of hydraulic control, or actuation, over mechanical or electrical, apart from the elimination of noise, to which Mr. Town referred, and the one or two comparisons he made showing the superiority of this actuation over mechanical. What I want to get at is this, is it the hydraulic control, or actuation which gives greater flexibility ? I would like the advantages to be defined briefly.

Another important omission, to my mind, is that of " maintenance cost." As I say, I have had little, or no experience, with hydraulically controlled tools, apart from presses, and therefore I have no idea as to how maintenance costs of such machines would compare with mechanical, or electrical, but I should imagine they are very high. No doubt Mr. Town could give some better comparison. If there are no advantages, or if maintenance costs are comparatively high, does the increased output obtainable justify the use of this system ? In one instance only, I think, did Mr. Town make a com-

parison of output. My recollection is that that was not an advantage, or the increase was not a great one.

Mr. H. C. Town: In regard to the question of "cushioning." Although we are taught that fluids are incompressible, it has been stated there is a slight cushioning effect, for it can be demonstrated on hydraulic presses where great pressure of several hundred tons can be exercised without any definite ram movement. As far as I can recollect, the percentage of that actual oil compressibility is about 0.35% for every 1,000 lb. per sq. in. pressure. In regard to the effect of this "cushioning"; the chief difficulty arises with motions such as you might require, we will say, for screw cutting. With the increase in the opposition, the tool slows down its traverse and this lost traverse is never regained. This may be, in some cases a big advantage, such as a cutting tool meeting an increased depth of cut. The tool simply slows up until the hard place is past, and then gains the normal traverse.

Regarding the feed motion of milling machines mentioned by Mr. Bingham. We get there, on some machines, a motion different to anything we have been led to expect in the machine tool world; that is the cutter approaching and springing away from the work. We have always tried on a mechanical machine to prevent this by eliminating backlash between the feed screw and nut; yet with a varying hydraulic traverse, as Mr. Bingham says, the effect on the work is no worse in regard to finish, and it is actually claimed that the cutter life is increased as much as 50% due to this flexible movement.

Regarding the definite advantages of hydraulic operation; I think they can be more or less summarised, as follows: There is a greater tool life, infinitely variable speed, quiet operation, and freedom from shock, the safety of machine is ensured by the valves. We eliminate many mechanical parts, that is, we do away with screws, gears, racks, clutches, and thereby, what we might term—the winding up of a mechanism which may comprise as many as twenty units—is altogether eliminated. There is no winding up of the parts, the connections are very few.

As to maintenance, as all the parts work in oil, which is the ideal fluid, long life is assured and the maintenance costs are not high. This development to machine tools is rather tending to take the machine tool engineer into the realms of a plumber, but it is mostly with the valve gear where difficulty is liable to occur, but, even then, as I said before, because we are working under ideal conditions, maintenance costs should not be high. I think that the output which is obtained does certainly justify the hydraulic operation, particularly on machines where the cutting forces are not very high, principally for grinding machines. Almost every grinding machine is

now fitted with hydraulic transmission and the same advantages can be obtained on practically all other types of machine tools.

Unfortunately, the Germans seem to be the pioneers of this development, and with a conservatism which is characteristic in this country—I think in regard to a lot of other things besides hydraulics—while I say at once that British machine tools are certainly as good, or better than, any others produced—in this subject we have just a tendency to lag behind the Germans until such time as we think we will go into the field, and thereby they get a bit of a start on us, and this applies particularly to milling machines and the like, where but little has been done in this country.

A MEMBER: Will Mr. Town give us his views on the relative merits of the rotary and the piston type of valve?

MR. TOWN: From my own experience I rather favour the rotary type of valve, but, in some cases the design of the machine more or less settles that. Both types can be satisfactorily used, and on the Fortuna-Werke Internal Grinding Machine they use a series of piston type valves which appear to work very satisfactorily, but to move a piston valve over much of a distance, you generally require a pilot valve, which, on being struck by a lever, actuates the second valve, the object being to shorten the movement to about ⅜ in. in most cases. The object, of course, is to get as near an instantaneous movement as you can get, and, in some cases, the piston valve, if it is supplemented with a pilot valve, will give a very rapid movement.

MR. AUSTIN: Mr. Town seemed to accept the suggestion of the feed on the milling machine being too intermittent, or jumpy. I saw a test on a Cincinnati milling machine, on which they had got a control box, and dial in front of it, and the feed on a heavy cut. yet the traverse was quite constant without springing of the cutter, You showed an illustration of the "Precimax" grinder on which the hydraulic motor drove the work. Would you mind telling me what is the advantage, and also any comparison you can give of the cost of the units?

MR. H. C. TOWN: In regard to the Cincinnati hydraulically operated machine, the purpose of the special valve was to eliminate the jerky feed and thereby, on that machine, you don't get the jerky movement, which is incidental to a machine which is not regulated on that system. Another feature is that you can control the viscosity and the regular feeding; you can fit a dial and actually give the feed in inches per minute, and guarantee that the machine will do, or rather will keep to the increases which are on the dial; otherwise the best you can do is to use a graduated dial and mark one end with "zero" and indicate the direction of the increase, and leave the machine more or less regulated by simply the switching of a lever on the dial, without giving a guarantee what the feed is.

In regard to the hydraulic motor on the grinding machine referred to. The feature is that you get an infinitely variable speed over a wide range, which is more effective on a large machine than a small one, and the hydraulic motor has its advantage over the electric motor, by giving increased torque with reduced speed. The alternative is to fit a variable speed electric motor. I cannot give any actual figures for the hydraulic motor, because it depends partly on the make of it, for one thing, but there is no doubt that, as made at the present time, the hydraulic drive is rather expensive. It may be, of course, that with the development of the drive, we can anticipate a fall in price, but, at the present time, the cost of these units does mitigate against their use. Take, for example, the number of hydraulic drives which are used on board ship, where several thousands of these units have been supplied. There is no reason, if the suppliers will go more thoroughly into the job, why they should not be able to bring the price down ; at the moment there is no doubt they are expensive.

Mr. Austin : I would like to refer again to this matter of " jumpy motion." After confirming my opinion that a jumpy motion of the milling machine was in no way detrimental to finish, Mr. Town quoted several advantages which that jumpy motion gave, including one, I think, of saving in tool cost. If that is the case, then I would ask if Cincinnati have intentionally sacrificed those advantages in the endeavour to eliminate the jumpy motion, and, if so, what compensations they have obtained in so doing ?

Mr. Town : The problem of milling machines is complicated if what we know as "climb " cutting is required. There is a development coming forward at present (another German development) in which machines are being designed to utilise " climb " cutting as against the normal cutting, where we always regard it that a cutter should commence to take a small chip, and should gradually work up to a large. With " climb " cutting the opposite takes place, and on a mechanical machine, if you are climb cutting, there is a tendency for a cutter to jump on top of the work. If you are going to cut only in one direction, it is easier to keep the cutter in a more or less locked position by oil pressure on both sides, equal pressure, than it is if a reverse direction is required, and I take it that it is this feature that the Cincinnati Milling Co. are getting : that is why they try to keep the machine as rigid as if it was on a mechanical feed. The Cincinnati Milling Co. themselves claim that the life of their cutters can be increased as much as 50%, so that they apparently appear to think they have lost nothing by means of this locked feed.

A Visitor : Has Mr. Town had any experience of " sludging " of oil at the bottom of tanks, or in enclosed circuits, or if he has had any experience of a need for flushing out of the oil ? It is

apparent that no matter how very often you change over, there is always sediment at the bottom of the tank.

MR. TOWN: I cannot say that I have had any experience in that direction. I can understand it is necessary to change the oil after a certain amount of life, but I have no recollection of any difficulties due to the oil. It is necessary that only a mineral oil should be used, and this is supposed to be, more or less, free from sediment, and I have had no experiences of difficulty due to the oil, in that manner.

A VISITOR: Has Mr. Town been able to come to any decision as to the maximum penetration of metal, either with drilling, or by other machines which hydraulic control has exposed? From my experience there is an absolute maximum penetration, or a maximum which can be used to remove metal, whereas with a mechanical machine you put in gears, the shear pin breaks, something happens, but with hydraulic control there is a point that you cannot pass as regards the cutting feed.

MR. TOWN: I have no actual experience of the tests of seeing what a machine will do in that direction. It does, however, open a very interesting field for research work. There is a feature, of course, that with the pressure gauge on the machine it should be comparatively easy to get tests in that direction but, as far as hydraulic operations go, the tendency has been more towards high speeds with comparatively light cuts, rather than against heavy cutting. The Wotan shaping machine can be cited as a case of very heavy cutting, and the machine will actually stop dead in the middle of a cut, and then start again almost simultaneously in a manner that I do not think you would get a mechanical shaper to do.

I did mention about the feature of a pressure gauge on the machine tools; this feature allows the operator to tell at a glance the condition of a cutting tool; the pressure rises and shows that the tool requires sharpening, and in regard to broaching machines using delicate broaches, the valve can be adjusted so that pressure cannot build up above a limited amount, and thereby the life of the tool is safeguarded.

In regard to the actual metal removal capacities of some of the hydraulic machines. There is an interesting field open there for experiments, which will be very useful to try, because with a relief valve, the machine is always safeguarded, for the valve simply opens, and you are not put to the expense of replacing shearing pins as on a mechanical machine. My experience of mechanical speed devices has not been altogether too happy in regard to shear pins; they generally shear and lock and the result is stripped gears, and if it is of the adjustable type it is difficult to prevent the operators tampering with those in some cases, but with fluid

pressure the machine is undoubtedly very much more safeguarded.

A MEMBER: Has any difficulty been occasioned by using unsuitable oil? In regard to our own experience using an Arctic grade of oil for hydraulic sawing machines, we found this to be most unsuitable when the oil temperature began to rise, and could not get satisfactory operation until we changed the grade of oil.

MR. TOWN: I think the speaker has been rather unfortunate in the supply of oil, because from experiments over quite a number of years we have found the Arctic grade of oil we obtain in Keighley, and obtain for some machines of my own design, to be the one that we can use for transmission. I do not know where the difficulty comes unless the speaker has been using too small an oil tank, and not making any provision for cooling the oil. On some of the machines it is customary, if the oil tank is small, to run the oil through a cooling radiator, on which water is spread by means of a small fan. The other method of course is to use a large enough oil tank so that a considerable amount of oil is kept in circulation.

I have a reference to the oil which is used in hydraulic couplings. That is the Shell BA.8, which is satisfactory for this purpose, and will run from about 4,000 to 6,000 hours at a temperature of 240° F.

A MEMBER: We used various kinds of oil but we have found that the DTB oil was the best for us, and I have since found out that is the oil which the Cincinnati recommend. The temperature of the oil is about 100°. We do not have any trouble at all; if we get complaints about hydraulics we straight away ask what sort of oil is being used?

MR. TOWN: I merely mentioned the Arctic oil, for we found from experience this to be the best where the difficulty has sometimes been with a sluggish feed in a very cold shop temperature, say on a Monday morning, and the difficulty has been that the machine will start sluggishly, and then after running a few hours, will run in a somewhat more satisfactory condition, but if provision is made with a large oil tank or a proper cooling device, I do not think any difficulty will occur with the Arctic make of oil at any reasonable temperature.

Discussion—Yorkshire Section.

MR. J. D. SCAIFE (Section President, in the Chair): Mr. Town himself has anticipated a few of the points that I wished to make. One was a mention of a planer, by Waldrich, a planer made in Germany. I remember that this firm made a hydraulic planer about three years ago, but I noticed when I paid a recent visit to Leipzig that this firm had gone back to a rack and pinion drive, because they found that the hydraulic principle of planing machines had not worked very well. I saw the hydraulic machine and noticed when the tool came against the work for fairly heavy cut the table stood momentarily and went away with a jerky motion. This fault is inevitable with hydraulic operation of heavy planing machines.

Mr. Town also mentioned hydraulic shapers. Making enquiries from a German friend of mine who is in close touch with this subject he tells me that one of the firms in Germany who make a hydraulic shaper and a mechanically operated shaper as well makes 19 mechanical units to every one hydraulic. It has been found on test that there is no advantage in hydraulically operated shaping machines over the conventional type with mechanical drive. These tests were made in a scientific manner, no advantage being found in hydraulic operation.

With regard to the hydraulic workhead on lathes, there is one important idea in a hydraulically operated workhead. On the lower speeds you get a higher torque and if you require infinitely variable speed control, this can only be obtained by hydraulic means. It is expensive, however, and something approaching its efficiency can be obtained by means of a variable speed electric motor used to bridge the gap between the speeds of the gear-box—that is by having say eight, 10, or 12 speeds, and then smoothing out the difference by a variable speed motor. Such a system is much cheaper than a hydraulically operated workhead.

MR. R. G. HEWITT: Mr. Town has proved himself to be an authority on hydraulics; he has not only shown us the developments in hydraulics but has also indicated future possibilities. To-night, with Mr. Town's help, we have "struck oil," and that right profitably. The application of hydraulic motors to lathe spindles is a favourite subject of mine, and although the critics of this form of drive lay stress on its high initial cost, their other argument regarding its loss in efficiency has just recently been exploded. In an article by Dr. Kronenberg, published recently in *The Machinist*, tests were taken on gear driven and hydraulically driven lathes. It was found that the hydraulic motor stood an overload of 90% without difficulty. The final conclusions were that the efficiency

of the hydraulic and mechanical drives are nearly equal at the higher speeds and normal loads. The difference in favour of the mechanical lathes is stated as being 1.7% at 480 r.p.m., 1.6% at 392 r.p.m., and only 1.4% at 300 r.p.m. At full load the hydraulic lathe works under the most favourable conditions because of its stepless speed variation, and has a chip-volume capacity 30% more than that of the mechanical machine, requiring however 38% more energy. One of the main factors in the widespread adoption of flexible vee and tex rope drives is the elimination of vibration at high speeds, and the increasing demand for flexible drives without vibration, is, in my opinion, the main argument whey the hydraulic motor will become more widely used. I should like to ask Mr. Town if any difficulty is experienced with the hydraulic motor when running at slow speeds? I believe it has been stated that there is a tendency for jerky movements.

Mr. Town: Commencing with the remarks of Mr. Scaife, who, of course, speaks as an authority and not as one of the scribes, we are bound to agree with him with regard to the remarks on the hydraulic planing machine. As I pointed out, the difficulties do increase very rapidly with the increase in the length of the stroke, but with the shaping machine it appears to me that there is a wide field for investigations. If I may return to the subject of the Waldrich planing machine, they have taken out yet another patent this year of another type of hydraulic drive, so that they appear to be still experimenting and anticipate that they will obtain eventually a satisfactory drive.

Shaping Machines. At the last Machine Tool Exhibition anyone who witnessed the "Wotan" machine would be bound to be impressed by the possibilities of heavy cutting demonstrated on that machine, such as that when the machine was stopped in the middle of a deep cut it would, without any run at all, start away without hesitation. The Americans are coming to the fore with one or two developments in regard to hydraulic shapers. The Rockford Co. are making claims that with their hydraulic shapers they get in nearly twice as many cuts as you can get in with the old mechanical drive—1,909, to be exact. This is a very big claim to make; it means a saving of twenty-eight minutes in every hour, a very big saving indeed.

With regard to the developments of hydraulic workheads for lathes, it appears to me that the time is coming when we will be driven to developing a suitable means of infinitely variable speed control over a wide speed range. If we go back a matter of six or seven years ago, a $12\frac{1}{2}$ in. centre lathe, we will say, had an adequate speed range with 10 to 360 r.p.m. which we might obtain with 18 speeds if we keep the German standard ratio of 1.26. We have now gone from 360 r.p.m. up to about 1,000 r.p.m. on the same machine.

I noticed last week one German lathe giving a speed range of from 10 to 1,180 r.p.m. with 27 to 36 speeds, depending, of course, on what you are prepared to pay for. Obviously, 27 to 36 speeds obtained by means of gears is getting out of range of convenient operation. It means a nest of operating levers, interlocking devices, and other mechanical means which, in my opinion, will soon make the cost of the lathe headstock equal to or more than that of the hydraulic drive.

There is one feature about the hydraulic drive which has not been mentioned regarding the strains taken off the electrical equipment. The cheapest types of motors can be used. This point might be emphasised ; it would help to bring down the cost of the drive. Mr. Scaife has pointed out one solution—to get part of the speed range mechanically and the other electrically. If we can do that, that part of the trouble is eliminated at once. In cases where we are compelled to use constant speed A.C. machines the point becomes more pronounced.

With regard to Mr. Hewitt, I think the last remark practically answers him about the simplicity of the electric drives. I must agree with him that it is not advisable to go below a certain range of speeds with the hydraulic motor, otherwise the motion is jerky. I should think that for a direct drive about 10 r.p.m. is as low as it is possible to get if any jerkless motion is desired. In the case of the Magdeburg lathe headstock with a double back gear you can get the speed down much lower.

Mr. Towler : When I started my career as a hydraulic engineer I think I saw that first diagram of Mr. Town's which showed a small ram and a large ram, and I thought " This is going to be very easy ; any darned fool can understand this." I continued in that line of hydraulics. But when you find the complicated diagrams we have seen this evening it gives one rather a shock. To many of us, probably more so to those people who are not really conversant with hydraulics, the diagrams are very complicated and very difficult to understand. It does seem to show that if the hydraulic drive has advanced so far as it has, in spite of these complications, there must be serious advantages in the technique, but at the same time I feel that a great deal of development may yet be called for in simplifying these valve arrangements. I feel sure there is room for simplification and that such development will eventually arrive. I feel also that the hydraulic machine tool is up against a very serious handicap in the scarcity of men who understand hydraulics of any sort. You come up against a complicated valve arrangement which needs almost a chess brain to understand it. I think, therefore, although I feel sure there is a very great future for the hydraulic machine tool in many lines, much must be done to simplify it before it can really come fully into its own field of use.

Mr. Town : In regard to the complications, it can be said at once that the machine tool probably provides the most complicated system of any type of machine, owing to the varied difficulties and wide speed range and the acccuracy which must be obtained in the finished product. The valves I have shown to-night may appear somewhat complicated in diagram form. Perhaps if they had been shown in parts they would appear to be somewhat simpler.

In regard to advantages, I might point out that the chief advantage is that we reduce in most cases the number of transmission elements such as gears, racks, pinions, screws, etc., with all their complications of bearings, and also we reduce nearly all possibilities of flexing in the transmission. Take a lathe, for example, from the drive to the final rack pinion there may be as many as, say, 20 or more pieces of mechanism ; " winding " as we term it, can take place, and although we try to overcome these difficulties in a gear drive by means of various slip devices, such are not altogether satisfactory, whereas with the hydraulic machine we can, as it were, go round corners, as I have pointed out. We cut out all possibilities of flexure and " winding " of the mechanism and therefore we secure more satisfactory and smooth transmission torque.

It is quite correct to say that there is probably a dearth of men at the present time accustomed to hydraulic machine tools. The machine tool designer, in many cases, has not taken very kindly to having this new field of operation thrust upon him, but a few years ago we got the same thing with electric drives. Even the use of ball bearings took a good number of years to get people to incorporate them. It is only a matter of time before workmen will get used to hydraulic systems and the advantages which accrue from them. Another superiority, of course, is that since all parts are working in oil, which is naturally the ideal condition, there is very little likelihood of wear or breakdown taking place and great advantages accrue thereby.

Mr. Arthur Sykes : I think we may be excused a little diffidence on this subject in the presence of an expert like Mr. Town and another expert like the President. We know that Keighley is the centre, in this part of the world at any rate, of the machine tool industry, and it is very gratifying to have gentlemen connected with our Institution like Mr. Town.

I was glad to hear from you, Mr. President, a word of warning about overdoing the application of a new invention. It very often occurs, and, from what you say, it has occurred to some extent in hydraulic mechanisms. I must say that I felt a little abashed, like Mr. Towler, with the complicated valve mechanisms which appear to be essential.

I was interested in the differential valve for controlling the effects of differential viscosities. Has it been found necessary to resort

to heating and cooling of the oil? I mean, when it cools down after a cold night or becomes too hot after a long period of running?

MR. TOWN: I am bound to agree that as with most new developments there is a tendency to incorporate hydraulic transmission in cases where it is not altogether the best means of obtaining a movement. I think that most of the machine tool makers have gone along various lines and have got it cut down to where it can be supplied successfully and where not. It is the same thing with all types of new developments. On the introduction of the grinding machine, for example, it was predicted in some quarters that the centre lathe would be done away with. We had the same thing with regard to the turret lathe eliminating the centre lathe. After a time they all find their proper level and only the most useful applications are retained.

In regard to heating and cooling the oil, there is a tendency, of course, for the temperature to rise during a day's work, and with rather curious results in some cases. With a constant delivery pump, as the viscosity becomes less the pump has a tendency to supply more oil and so the feed rate increases. Alternatively, with a sluggish or cold oil the feed is reduced. This is a reason for the elaborate valve mechanism on the "Cincinnati" machine, because they wish to place on the machine a dial giving definite feed in inches per minute. If the viscosity should vary throughout the day you cannot be tied down to that because the oil flow and the feed will vary slightly at the different positions of the control lever. The most usual method of guarding against this heating and cooling is to use a suitable grade of oil which will not become unduly sluggish in cold weather, and again, if the oil temperature tends to rise unduly, either to supply a large tank so that a large quantity of oil is always kept in circulation, or to provide an oil cooling radiator, as used on the "Precimax" machine, whereby a normal temperature is maintained throughout the day. In some of the older type machines it was common to use carbon filament lamps for oil heating but a crude method like that would not be used to-day. If we go back on some of the earliest lathes, notably in some of the gunshops at Elswick, that method was used to raise the temperature of the oil on cold mornings.

MR. J. HORN: Can Mr. Town tell us if the hydraulic principle has been used in any other machines than light machine tools or presses? I should also like to know how rapidly such movements can be used on light mechanisms?

MR. TOWN: In regard to other types of machines than machine tools and presses, I think at the present time there is only a very

limited field, although in regard to hydraulic drives they can be found on practically every type of machine, cranes, winches, and textile machinery drives, but not so much for feeding motions as for rotary motions. It is this that makes me think that the machine tool makers are perhaps a little bit slow in going more fully into the matter of hydraulic drives. The Variable Speed Gear Co. alone have supplied several thousands of these drives, largely on board ship, for operations such as lifting shells, gun loading, and steering purposes.

Now in regard to the maximum speed of operation, I was asked by a local firm to see if it was possible to obtain a speed of 400 strokes per minute for a textile machine. I think that is excessive. I do not know what the maximum is but some of the reversals are instantaneous, and if you attempt to get up to 400 r.p.m. I should think you would have rather serious trouble through overheating of the oil. I cannot give any definite figures or the maximum figures that have been obtained. Mr. Scaife may be able to help in that direction.

Mr. Scaife : I haven't any exact case in mind. If the load were light there would be no difficulty. On Mr. Town's own figures of 8 ft. per second, that brings it well within 400.

Mr. Towler : Ram pumps are operated up to 2,000 r.p.m., but I should think 2,000 is about the limit, and they are only very short stroke pumps. People like the Constantinesco Gear Co. made during the war, a mechanism for firing machine gun bullets through the propeller blades. The speed depends upon the length of the stroke.

Mr. Scaife : The Constantinesco Gear Co.'s arrangement was not rotation. It was the oscillatory motion of a little ram which controlled the trigger.

A Visitor : I have listened with very great interest to Mr. Town's lecture. I am surprised there has been no mention of a form of transmission which is made in this city. This transmission has a pump and swash-plate rotary transmission unit.

Mr. Taylor : Mr. Town has mentioned the uses to which hydraulic power can be applied in rotational and straight line movements. He has pointed out that a constant rate of propulsion is required. Hydraulic motion is not very suitable, because variable resistances result in a variable rate of progress of the saddle or whatever it may be. Hydraulic transmission is not suitable for conditions where the resistance to motion varies very widely. I have seen in one of the technical papers a description of a machine developed by the Cincinnati Co. in the form of a die sinking machine.

A stylus controls the motion of a cutter, so that it travels in harmony with the stylus. I should like to know whether Mr. Town thinks such a device can produce really accurately, or whether there is a distortion in the job being produced between cuts and the work? It would seem to me that a device of that sort could not be extremely accurate. Can Mr. Town throw any light on this subject?

Mr. Town: In regard to a constant feed rate not being available with hydraulic means, this is quite correct, and in some cases it is an advantageous feature, such as in the case of a cutting tool meeting an increased resistance, or a hard place in a casting, the feeding rate slows down. But as the lost traverse is never regained it is not suitable for screw cutting or any other motion where accurate relation must be maintained. Although die-heads have been used operated hydraulically it is not feasible to cut screws.

In regard to the relay system as used on the "Cincinnati" machine it would appear that there is bound to be a certain amount of variation in the feed rate; although the differential valve is supposed to check that, there is bound to be just a small time lag, but they do claim that with the dial on the machine they can guarantee that the feed on the dial will be substantially maintained. From some experiments on attempting to cut screws it would appear that about 1% was the closest that was achieved in the accuracy in the screws obtained.

Mr. R. J. Mitchell: I have listened with great interest to Mr. Town's admirable paper, and especially to the reference to difficulties in the application of hydraulic power on heavy planing machines. I would like Mr. Town to tell us what would be the objection to designing a closed circuit system for operating planing machines consisting, in effect, of a duplex cylinder and piston with the pump and connecting pipe lines in a closed circuit operated at the end of each stroke by reversing the electric motor which drives the pump. It seems to me that you would get some possibility of advantage there arising from the imcompressibility of the working fluid. Could not this factor be utilised to absorb the shock when the resistance to motion became unduly high, and then collapsed suddenly as would occur in overcoming an obstacle such as a hard spot? The arrangement might be likened to an endless hydraulic chain.

Mr. Town: Mr. Mitchell's is an interesting suggestion but difficulty arises from the very high table speeds required at the

present day. A few years ago such a system could probably have been worked, but enormous stresses are set up in the pipe lines when trying to reverse a moving column of oil rapidly. Stresses are set up of extreme severity which would be liable to cause the equivalent of water hammer in the pipes. Such stresses have been proved to be somewhat dangerous to the transmission and all that you get is a very sluggish reversal in place of the instantaneous reversal which you might expect. The system put forward by Mr. Mitchell has been used as a fluid linkage for operating the slides on a turret lathe in place of the usual cams and appears to work excellently in that connection. There is another alternative for planing machines and shaping machines and that is to retain the usual mechanical elements and introduce the hydraulic motor as a speed changing device in between the prime mover and the mechanical elements. One of the first machines to utilise this was a "Bateman" planing machine which was converted and used a variable speed gear drive and appeared to work fairly satisfactory at that time—1912—but, of course, the speed ranges have increased so much to-day that what satisfaction they would get to-day I do not know. A German shaping machine does use that method to-day, and they get from 10 to 90 strokes per minute and a pressure of about 9 tons on the ram. They use a hydraulic drive working at pressures of 900 lb. per sq. in.

Mr. Scaife: Whilst we have members in our ranks of the calibre of Mr. Town I think we can ensure a very long life for this institution. I think it is exceedingly fine that we have members like Mr. Town in our midst. It gives me very great pleasure indeed to propose a vote of thanks to Mr. Town for his excellent paper.

Mr. W. Grover (Vice-President): I am very glad to second that resolution. I join those who perhaps feel a little fear of those valves regarding leakages when you are dealing with a fluid in a system involving valve glands. They appear very formidable to the ordinary layman. What is going to happen? I suppose they *never* have leakages! I have heard it said about the fluid brake "Of course, if that thing leaks you are done." I am bound to say that on one occasion mine did leak! One of the union joints had to be tightened up. When you come to the fluid fly-wheel it is another matter: I am also very interested in that, but I have no fear of leakage there, all the pressure when full out being centrifugally taken by the casing, the pressure does not come on a joint. There

is only a gland in operation when the gear is practically at rest or full slip.

MR. TOWN: I would like to thank you for a very patient hearing. With regard to the remarks on leakage by Mr. Grover, one of the advantages in a certain amount of leakage is that the machine automatically lubricates itself, thus saving special lubricating arrangements. If the leakage becomes unduly great he is also safeguarded by mounting the units in or on the oil tank and the pipes within the machine bed, and therefore the customer is unaware that leakage takes place.

ANNUAL REPORT

and

ACCOUNTS

For the Year ended June 30, 1937.

To be presented at the

ANNUAL GENERAL MEETING

October 15, 1937,

At Institution Headquarters,
British Industries House,
Marble Arch, London, W.1,
at 7-15 p.m.

THE INSTITUTION OF PRODUCTION ENGINEERS.

BALANCE SHEET AS AT 30TH JUNE, 1937.

LIABILITIES.	£	s.	d.	£	s.	d.
SUNDRY CREDITORS				93	16	0
SUBSCRIPTIONS RECEIVED IN ADVANCE				41	15	6
Lord Austin Prize Fund ...	£52	10	0			
Building Fund	182	10	0			
Hutchinson Memorial Fund	37	12	6			
				272	12	6
INCOME AND EXPENDITURE ACCOUNT:						
Balance at 1st July, 1936	£1910	3	3			
Add Excess of Income over Expenditure for the year	486	17	2			
				2397	0	5

			ASSETS.	£	s.	d.	£	s.	d.
			FURNITURE, FITTINGS, AND PLANT at cost *less* amount written off:						
			Balance at 1st July, 1936	£516	18	7			
			Additions during the year	149	15	7			
				666	14	2			
			Less amount written off	147	19	0			
							518	15	2
			INVESTMENTS at cost:						
£588	3	0	Plymouth Corp. 3% Stock, 1956	591	18	6			
£625	17	1	Ayr C.C. 3% Stock, 1956 ...	630	9	6			
£600	0	0	London Cnty. Consolid. 3% Stock, 1956-61	603	17	0			
1814	0	1		1826	5	0			

	£	s.	d.		£	s.	d.	£	s.	d.
				Less Profit on re-investment	200	9	2	1625	15	10
				(Market Value £1726 6s. 4d.)						
				SUNDRY DEBTORS				227	13	5
				SUBSCRIPTIONS IN ARREAR, *not valued*...				—		
				CASH :						
				At Bank	392	12	1			
				In Hand	40	7	11	433	0	0
	£2805	4	5					£2805	4	5

(*Signed*) J. H. BINGHAM, *Chairman of Council.*

(*Signed*) WALTER G. KENT, *Chairman, Finance Committee.*

(*Signed*) R. HAZLETON, *General Secretary and Treasurer.*

AUDITORS' REPORT.—We have audited the above Balance Sheet dated 30th June, 1937, and we have obtained all the information and explanations we have required. In our opinion such Balance Sheet is properly drawn up so as to exhibit a true and correct view of the state of the Institution's affairs according to the best of our information and the explanations given us and as shown by the books of the Institution.

Aldwych House,
London, W.C.2.
7th September, 1937.

(*Signed*) GIBSON, APPLEBY & Co., Auditors.
Chartered Accountants

THE INSTITUTION OF PRODUCTION ENGINEERS.

INCOME AND EXPENDITURE ACCOUNT FOR THE YEAR ENDED 30th JUNE, 1937.

DR.

	£	s.	d.	£	s.	d.
To Salaries...				1024	5	11
,, Rent, Lighting, Heating, and Cleaning				303	0	9
,, Local Section Expenses				331	16	5
,, Printing, Postages, Stationery, Telephone, and Certificates ...				275	6	9
,, Printing, Postages, Stationery, and Editing—Journal				820	0	3
,, Staff Travelling and Expenses of General Meetings				100	2	9
,, Professional Charges, Insurances, and Income Tax				136	15	1
,, Donations to :						
International Congress for Testing Materials	25	0	0			
British Management Council	5	5	0			
				30	5	0

CR.

	£	s.	d.	£	s.	d.
By Subscriptions received :						
Current...	2745	0	0			
Arrears...	85	9	0			
				2830	9	0
,, Interest on Investments				48	6	7
,, Receipts from Sales and Advertisements of Journal				822	4	7
,, Donation from Lord Austin ...				52	10	0

	£	s.	d.		£	s.	d.
,, Library...	7	10	9				
,, Examinations	34	18	10				
,, Annual Dinner, less receipts ...	21	6	1				
,, Miscellaneous	33	5	5				
,, Amount written off Furniture, Fittings, and Plant	147	19	0				
,, Balance, being Excess of Income over Expenditure	486	17	2				
	£3753	10	2		£3753	10	2

ANNUAL REPORT FOR 1936-37.

To be presented by the Council to the Annual General Meeting, London, 15*th October*, 1936.

Membership.

The membership at the end of June, 1937, was as follows :—

Honorary Members	5
Ordinary Members	537
Associates	32
Associate Members	587
Graduates	284
Affiliates not already included in other grades...	26
Affiliated firms	35
	1,506

Two hundred and three new members were added to the Register during the year. Six members died, 21 resigned, and 30 lapsed. The members whose deaths have to be recorded with much regret were Messrs. L. Riley, B. Jackson, E. G. Latty, P. R. Painton, C. H. Sanders, and E. A. Swift.

Finance.

As a result of the satisfactory surplus shown in the accounts, a sum of over £460 has become available to add to the Institution's investments, all of which are now in Corporation Stocks. The Council has to thank Lord Austin for a second donation of 50 guineas.

Awards.

The winner of the Lord Austin prize for the best attainments at the 1937 Graduateship Examination was Mr. R. A. Cox, Birmingham.

The medal for the best paper by a member last session has been awarded to Mr. H. C. Town for his paper on " Hydraulic operations of machines." For the previous session, Mr. W. Puckey, President of the London Section, gained the award for his paper on " Personal Problems of Management."

The Hutchinson Memorial Medal for the best paper by a Graduate, 1936-37 session, has been won by Mr. A. E. N. Bolton, late of London, now a member of the Glasgow Section, for his paper on " Stressed Wood Construction." The first award of this medal was made the previous session to Mr. Harry Shaw, Manchester Section, for his paper on " Recent Developments in the Measurement and Control of Surface Roughness."

New South Wales Centre, Sydney Section.

The first overseas Section of the Institution is in process of establishment at Sydney, Australia, where a meeting was held in June, 1937, to further the project.

Lectures and other Activities.

The quarterly reports submitted to the Council from the various Section Committees covering the work of the past session have been very satisfactory, particularly as regards the lectures and discussions. Attendances at meetings in most Sections were larger than the previous year. It is gratifying to find that the three junior Sections at Birmingham, Coventry, and London are making steady progress. A new departure last session was the holding of four Regional Conferences for discussing Section programmes. It has been decided to hold similar Conferences next session.

Thanks to Lecturers.

Once again the Council, on behalf of the Institution, wishes to thank lecturers, both members and non-members, for their services.

Gift to the Institution.

Thanks have been conveyed to Mr. Bingham, Chairman of Council, for the gift of an oil painting of James Watt, a reproduction of which, in colour, was executed by our Printing Department and published in *The Journal*.

Donations to other Organisations.

A grant of £25 was made during the year towards the expenses of holding in London the recent International Congress for Testing Materials, and a subscription of 5 guineas per annum is being paid

to the British Management Council which was set up under the chairmanship of Lord Leverhulme in January, 1937.

Standardisation.

The report of the Institution's Standards Committee, published in March, 1937, shows that keen interest is being taken in the work of engineering standardisation. Very close co-operation with the British Standards Institution is being maintained, and the Chairman of our Standards Committee, Mr. H. A. Hartley, now represents the Institution on the Mechanical Industries Committee of that Organisation.

Mr. W. G. Groocock.

After several years of valuable work, Mr. Groocock has retired from the Council of the Institution and takes with him the good wishes of all his late colleagues.

The President.

On behalf of the members, the Council tenders to Lord Sempill its grateful thanks for all that he has done for the Institution in his two strenuous years of office as President, during which his leadership has added much to its strength and prestige.